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Automotive Stirling Reference Engine Design Report

Stirling Engine Systems Division
Mechanical Technology Incorporated

June 1981

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract DEN 3-32

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U.S. DEPARTMENT OF ENERGY
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FOREWORD

This design report describes the Automotive Stirling Reference Engine System Design (RESD) and is submitted to DOE/NASA in support of the requirements of the NASA Contract: DEN 3-32.

SECTION 1.0 INTRODUCTION

1.0 INTRODUCTION

The Reference Engine System Design (RESD) is defined as the best engine design generated at any given time within the Automotive Stirling Engine Program that will provide the best possible fuel economy, and that will also meet or exceed all other program objectives. It was designed to meet all the requirements of a reference vehicle, which is representative of the class of vehicles for which the engine might first be produced. The vehicle selected was an X-body Pontiac Phoenix. The RESD utilizes all new technology that can reasonably be expected to be developed by 1984 and which is judged to provide significant improvement relative to the risk and cost of its development.

The RESD presented in this report has evolved from a series of design reviews starting in February, 1979. At that time, a conceptual study was made to evaluate possible options for the RESD. The conclusion was made to design a "U" drive system using an involute heater head with outer canister regenerators and coolers. This concept was then given more design attention to further reflect development experience and technology growth expected within the ASE program. An initial Reference Engine Design Review was held in January, 1980. However, it was felt that the engine design could be further improved and, therefore, updates of the RESD were presented in May, 1980 and March, 1981. This report describes the final engine design.

The RESD serves as a focal point to guide all component, subsystem and system development in the program. The Mod I and Mod II engine systems are experimental versions of this design. The Mod II is expected to meet the Final Program Objectives which are to develop, by September, 1984, an Automotive Stirling Engine System which, when installed in a late-model production vehicle, will:

1. Using EPA test procedures, demonstrate at least a 30% improvement in combined metro-highway fuel economy over that of a comparable production vehicle. The comparison production vehicle will be powered by a conventional spark ignition engine. Both the Automotive Stirling and spark ignition engine systems will be installed in identical model vehicles* and will give substantially the same overall vehicle drivability and performance. The improved fuel economy will be based on unleaded gasoline of the same energy content (Btu/gal).

   *It is intended that identical model vehicles be used for the comparison. However, a difference in inertia weight between the two vehicles is acceptable if the difference results from the substitution of the Automotive Stirling Engine System for the spark ignition powertrain system. The transmission, torque converter, and drivetrain may also differ in order to take advantage of Stirling engine characteristics.

2. Show the potential of gaseous emissions and particulate levels less than the following: $NO_x = 0.4$, HC = .41, CO = 3.4 gm/mile and a total particulate level of 0.2 gm/mile after 50,000 miles.

   The potential need not be shown by actual 50,000 mile tests, but can be shown by Contractor projections based on available engine, vehicle, and component test data and emissions and particulate measurements taken at EPA using the same fuel as used for the EPA fuel economy measurements.

In addition to the above objectives to be demonstrated quantitatively, the following will be considered system design objectives of the Mod II:

1. Ability to use a broad range of liquid fuels from many sources including coal and shale oil. This objective will be investigated initially in the combustor development effort and later in engine and vehicle testing. The candidate alternative fuels and their characteristics to be considered in this Program will be identified based on the DOE Alternative Fuels effort. Until these specific fuels and their characteristics are identified for inclusion in the Program, diesel fuel, gasohol, kerosene, and No. 2 heating oil will be used as a representative range of alternate fuels. Engine tests with the alternate fuels will not be initiated for the ASE Mod I and ASE Mod II engines until satisfactory operation, performance, and emissions have been achieved with the baseline fuel: unleaded gasoline. Testing will then be conducted with the selected alternative fuels to determine the extent of any detrimental effects on engine operation, performance, emissions, or fuel economy and to determine the degree of modifications or adjustments that might be required in switching from one fuel to another.

2. Reliability and life comparable with powertrains currently on the market.

3. A competitive initial cost and a competitive life-cycle cost with a comparable conventionally-powered automotive vehicle.

4. Acceleration suitable for safety and consumer considerations.

5. Noise and safety characteristics that meet the currently legislated or projected Federal Standards for 1984.

In the process of designing the Reference Engine, consideration was given to the performance and reliability of critical parts similar to the Mod I design, and to the test results from the component development efforts. The engine was optimized for the EPA automotive driving cycle. This optimization process, which was made for part power operation (rather than maximum power), resulted in: 1) reduced size Stirling cycle components, 2) maximum efficiency near the part power average operating point, and 3) sizing of auxiliaries to match the demands of the driving cycle. As a result, the regenerator cross sectional area and the gas cooler area and length became smaller relative to earlier engine designs, and the heater contained fewer heater tubes. This resulted in a heater cage outer diameter which was compact when compared to the baseline P-40 engine.

The following is a summary of the vehicle for which the RESD was designed:

1984 Pontiac Phoenix X-body

Mid-size

5 passenger

Curb weight	2870 lb
EPA inertial weight	3125 lb
Transmission	4-speed automatic with lock up in 3rd and 4th gears
Accessories	Power steering/brakes, air conditioning
Fuel	Unleaded gasoline (113,525 Btu/gal)

The following is a summary of the RESD design:

Height	700 mm
From shaft	490 mm
Width	690 mm

Length	850 mm
Weight	397 lb (180 kg)
Power	80.5 hp (60 kW)
Power Density	4.9 lb/hp

Maximum Power Point

Power	60.1 kW
Speed	4000 rpm
Working Gas Pressure	15 MPa
Efficiency	34.2 %

Maximum Efficiency Point

Power	22.1 kW
Speed	1100 rpm
Working Gas Pressure	15 MPa
Efficiency	43.5 %

Part Power Optimization Point

Power	12.2 kW
Speed	2000 rpm
Working Gas Pressure	5 MPa
Efficiency	37.7 %

When installed in the Reference Vehicle, the system performance predictions are:

- Fuel Economy (combined EPA cycle) — 41.1 mpg with unleaded gasoline (113,525 Btu/gal.)

  47.3 mpg with #2 diesel fuel (130,456 Btu/gal.)

- Acceleration (0-60 mph) 15.0 sec
  Acceleration (50-70 mph) 10.5 sec.
  Acceleration (0-100 ft.) 4.5 sec.

- Emissions (g/mile)

NO_x	< 0.4
HC	< 0.41
CO	< 3.4
Total Particulates	< 0.2

The engine system is broken down into five major subsystems:

- External Heat System;
- Hot Engine System;
- Cold Engine System;
- Engine Drive System;
- Control System Plus Auxiliaries.

Figure 1.0-1 shows the area of the engine where each system is located, with the exception of the Control System and Auxiliaries. Figure 1.0-2 identifies the major components within each system.

The External Heat System is comprised of the combustor or heat generating system, the preheater and housing, fuel injectors, vaporization chamber and flame tube. The system utilizes air-cooled thin insulation and is designed for ease of fabrication and simplified assembly; it also has a removable preheater matrix for easier maintenance. To control exhaust emissions, the External Heat System incorporates a Combustion Gas Recirculation (CGR) system which utilizes straight guide vanes with ejector nozzles. The CGR was designed for a constant air excess factor of 1.1 over the entire engine operating range. This design resulted in a high External Heat System

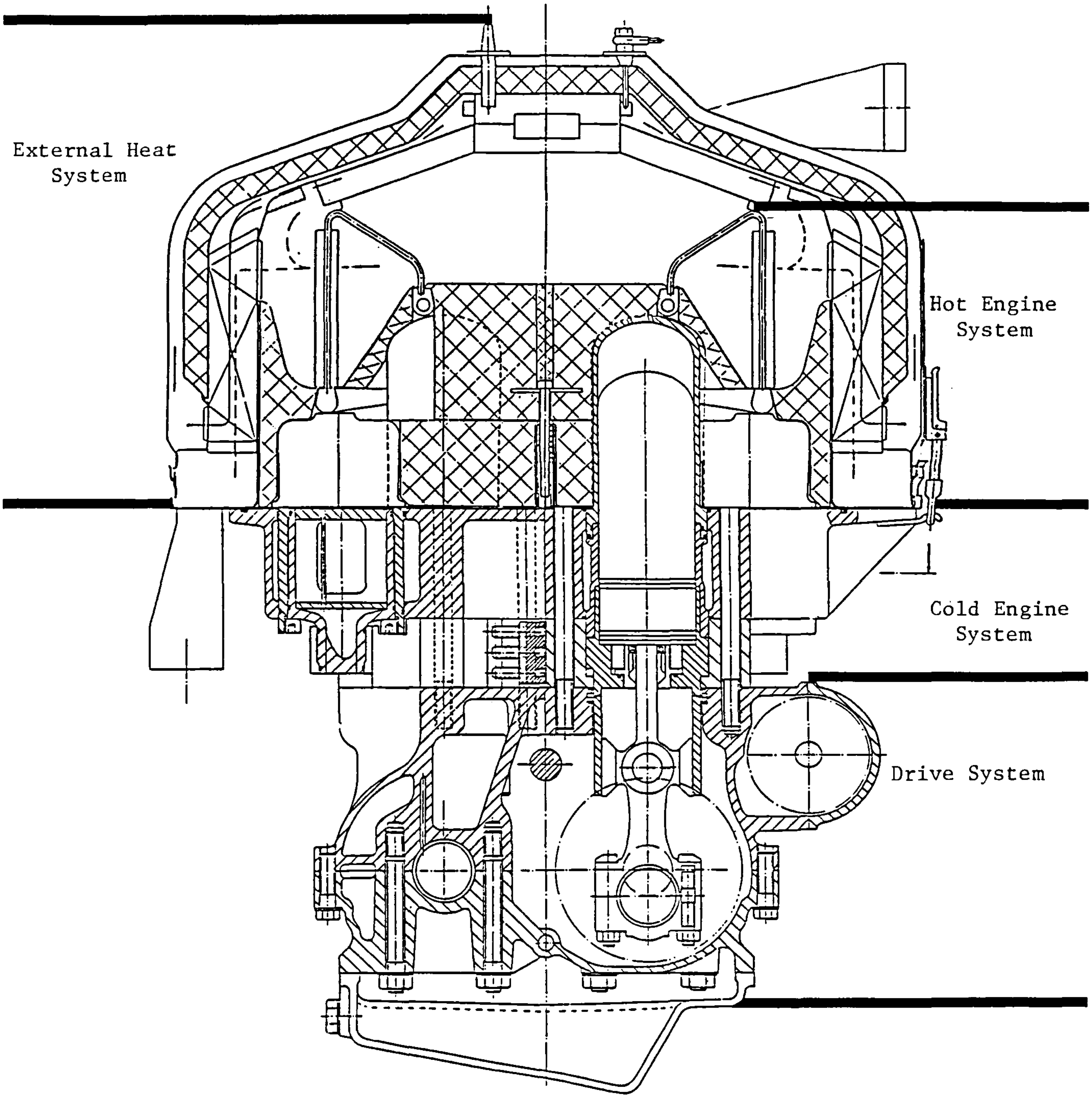

Figure 1.0-1 Reference Engine Design

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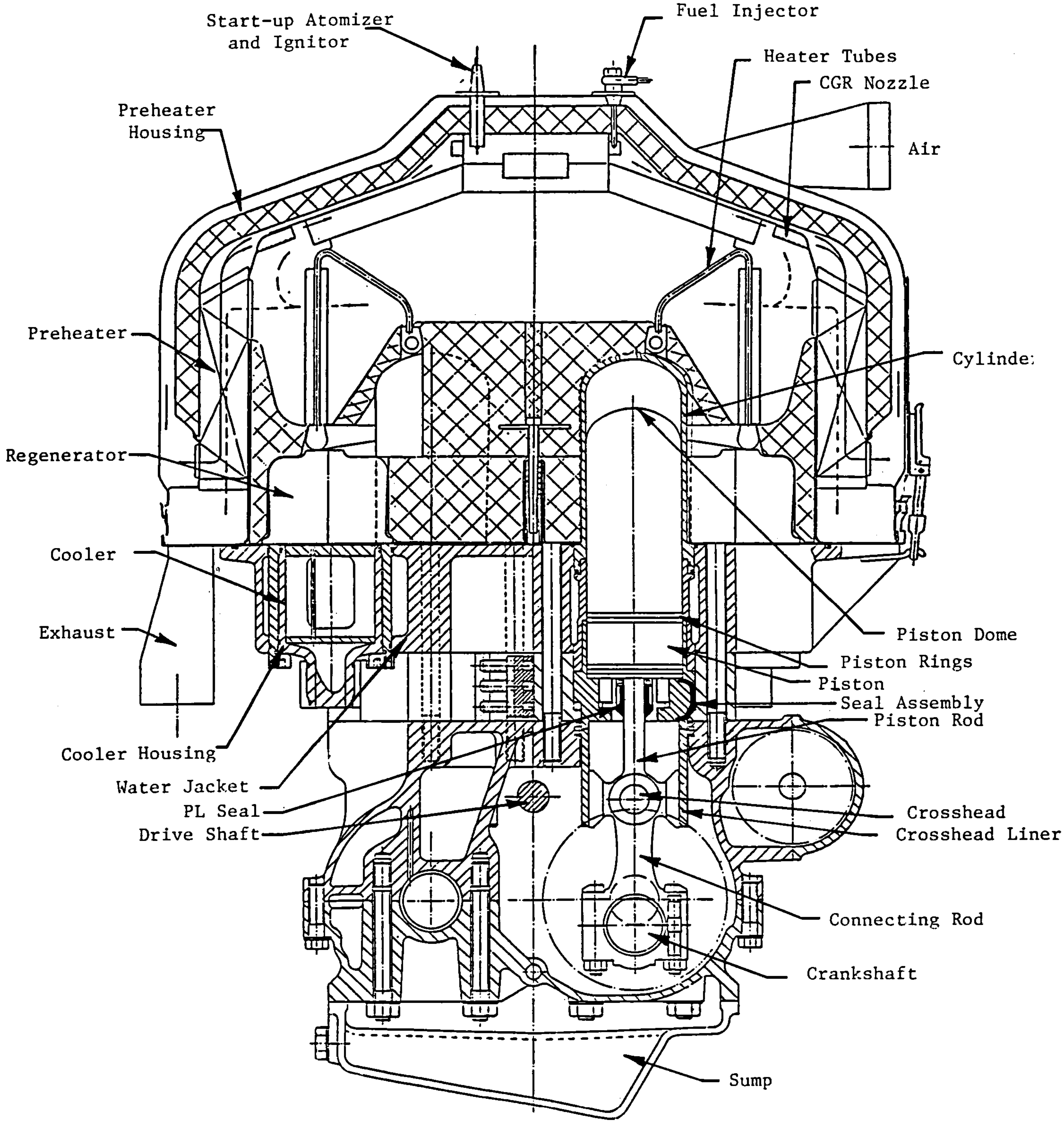

Figure 1.0-2 Major Components in Reference Engine Design

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efficiency, while it maintained a preheater plate cold-end temperature safely above the acid dew point (to avoid sulfuric acid deposits clogging the exhaust gas passages of the preheater).

The Hot Engine System is comprised of a four cylinder heater head, and four canister regenerators and coolers. The heater head is of involute construction with constant tube gaps in each of the two tube rows; it is designed to function with a heater head metal temperature of 820°C.

The regenerator and cooler housings are located immediately below the pilot-circle of the second row of heater tubes, to ensure even flow distribution. Their wall thickness and height were designed to minimize conduction losses yet meet the required stress criteria.

The regenerators consist of a stack of several hundred metal gauze disks which are sintered to prevent shifting and porosity changes within its cylindrical casing. The coolers, which are located concentrically below the regenerators, are constructed of many vertical seamless tubes, brazed to the top and bottom of their casings.

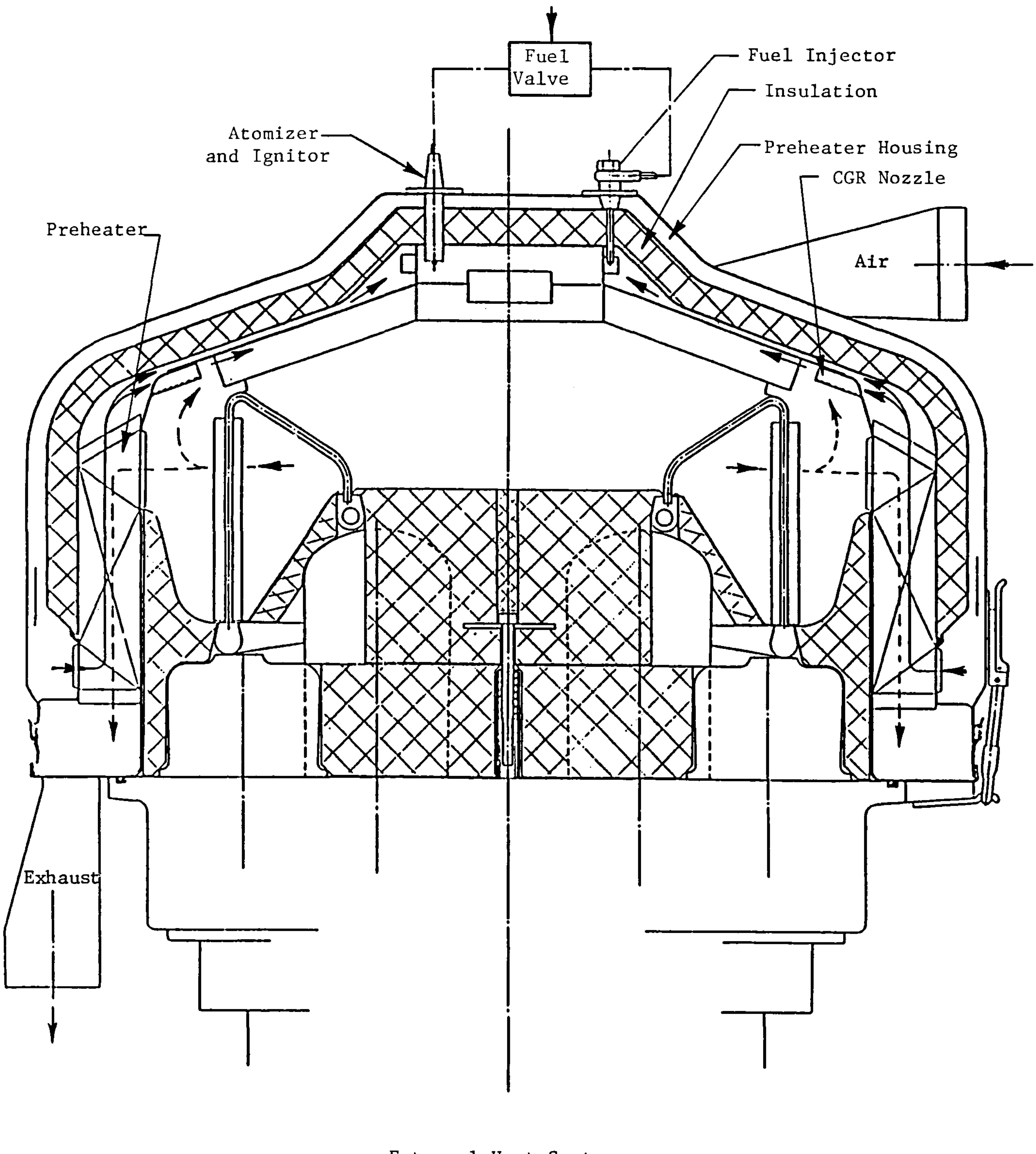
The Cold Engine System is comprised of the water jacket, the seal system (which includes the piston rings and Pumping Leningrader (PL) seals), pistons, and the piston rod assembly. The aluminum water jacket has split water flow passages of equal flow distribution and incorporates integral cylinder liners and crosshead guides to improve the alignment of the pistons. The piston rod seal system is located well up into the bottom of the piston to permit a reduction in overall engine height.

The Engine Drive System includes the crankcase assembly and power take-off assembly. The RESD is a 'U' drive with two crankshafts geared to one center

power shaft. It has an all-gear synchronization and is matched to the drive system of the X-body vehicle through the use of a speed increaser, which provides an output shaft speed of 1.4 times the engine speed. A separate balance shaft running at engine speed is used to maintain engine balance.

The Control System includes the mean pressure power control system, the air/fuel system, and electronic controls. Auxiliaries include the combustion air blower, the blower starter, the engine starter, the cooling fan, the alternator, and the water and fuel pumps. The auxiliary and power control arrangement is governed by the basic engine profile. Attempts were made, however, to achieve compact but easily serviceable locations and as few parts as possible.

SECTION 2.0 EXTERNAL HEAT SYSTEM



External Heat System

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2.0 EXTERNAL HEAT SYSTEM

2.1 Combustion Gas Recirculation System (CGR)

Combustion Gas Recirculation (CGR) was favored over Exhaust Gas Recirculation (EGR), which was used in the P-40 engine. The major advantage of the CGR system over the EGR system was the increased efficiency of the external heat exchanger system. Calculations performed for the Mod I have shown a gain in mileage of about 1 mpg in the CVS cycle.

The CGR system consists of a number of channels divided by straight partition walls (vanes). In front of each channel, a sheet metal ejector is located through which the preheated air flows. The air pressure after it passes through the ejector is below that of the combustion gas after it passes through the heater; this causes part of the combustion gas to mix and flow with the preheated air into the combustor.

In order to reach as high a fuel economy as possible, the following design was used: 1) a Combustion Gas Recirculation system (CGR), combined with fuel injection into preheated air in a vaporization chamber; 2) a folded counter-flow recuperative-type preheater with combustion air entering the preheater through a channel between the outer insulation and the shell.

It was found that a higher percentage of CGR could be obtained by using straight vanes rather than involute-shaped guide vanes, at the same pressure drop. The ejector area was designed to give around 50% CGR, which was sufficient for NO_x control. The air excess factor was designed to be 1.1 over the entire load range.

The air excess factor, percent CGR, and total pressure drop are shown in Figures 2.1-1, 2.1-2, and 2.1-3. The resulting efficiency of the External Heat System, therefore, was higher than the P-40 or the Mod I; efficiency is shown as a function of fuel flow in Figure 2.1-4. The design still maintains a preheater plate cold-end temperature which is safely above the acid dew point.

2.2 Fuel Injection

In the baseline engine (P-40) and Mod I combustion systems, fuel atomization was accomplished by pressurized atomizer air, which required a separate air compressor and adversely affected the efficiency of the External Heat Exchange System, especially at low loads. The External Heating System design offered the possibility of eliminating the air atomizer.

In the RESD, fuel is injected into a circular vaporization chamber, which is located on top of the combustor. The vaporization chamber has two opposing tangential inlets through which preheated air flows. A fuel injector is located in each air inlet. The flow area of the air inlets to the vaporization chamber was sized to accommodate about 20 percent of the total air from the preheater.

During start-up operation, the fuel flow required is about 0.6 g/s, which is roughly one-fourth of the fuel flow used, at start-up with systems using air atomization approaches. An igniter, located downstream from one of the fuel nozzles, ignites the air/fuel mixture.

When the walls of the vaporization chamber have reached a temperature high enough to vaporize the fuel, the fuel flow is increased and a

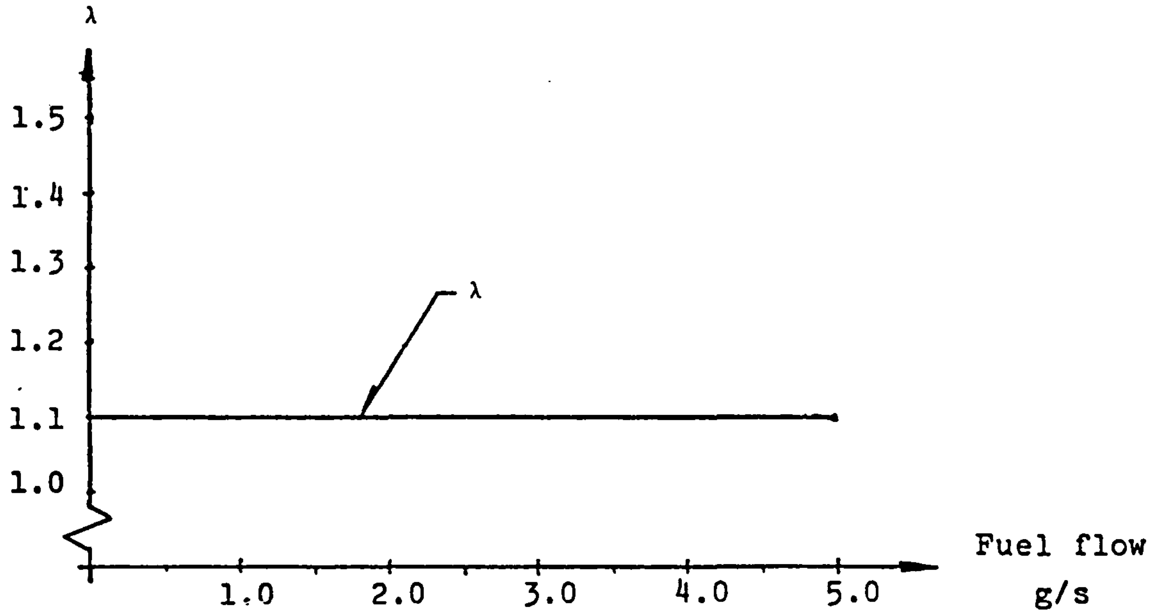


Figure 2.1-1 Air Excess Factor (λ) versus Fuel Flow

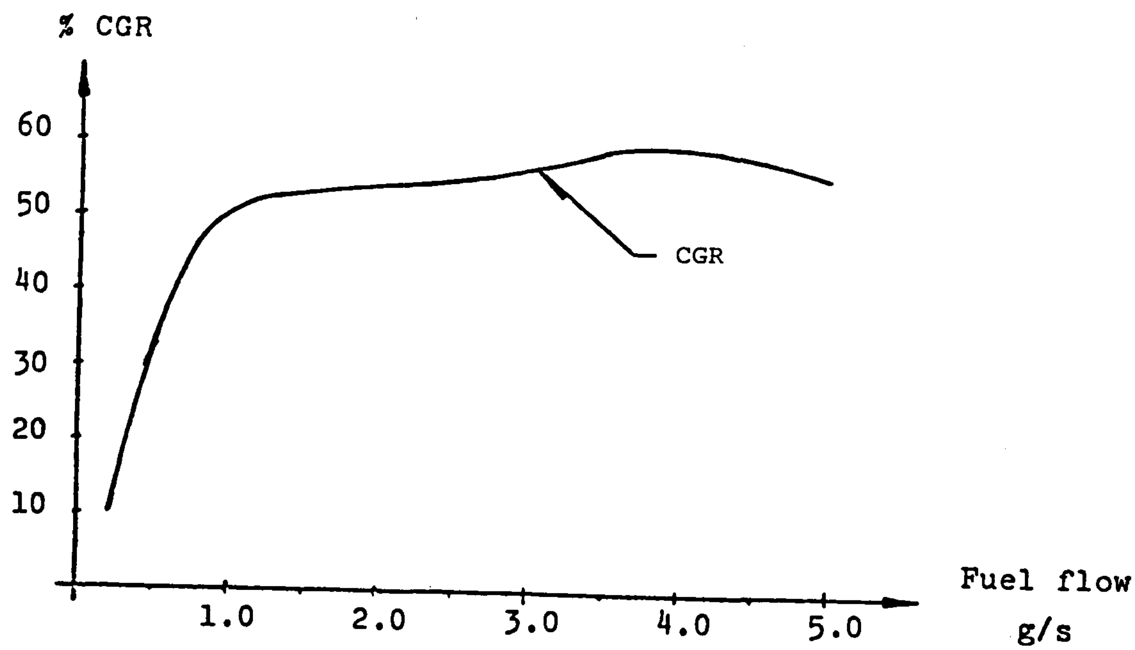


Figure 2.1-2 Percent CGR Versus Fuel Flow

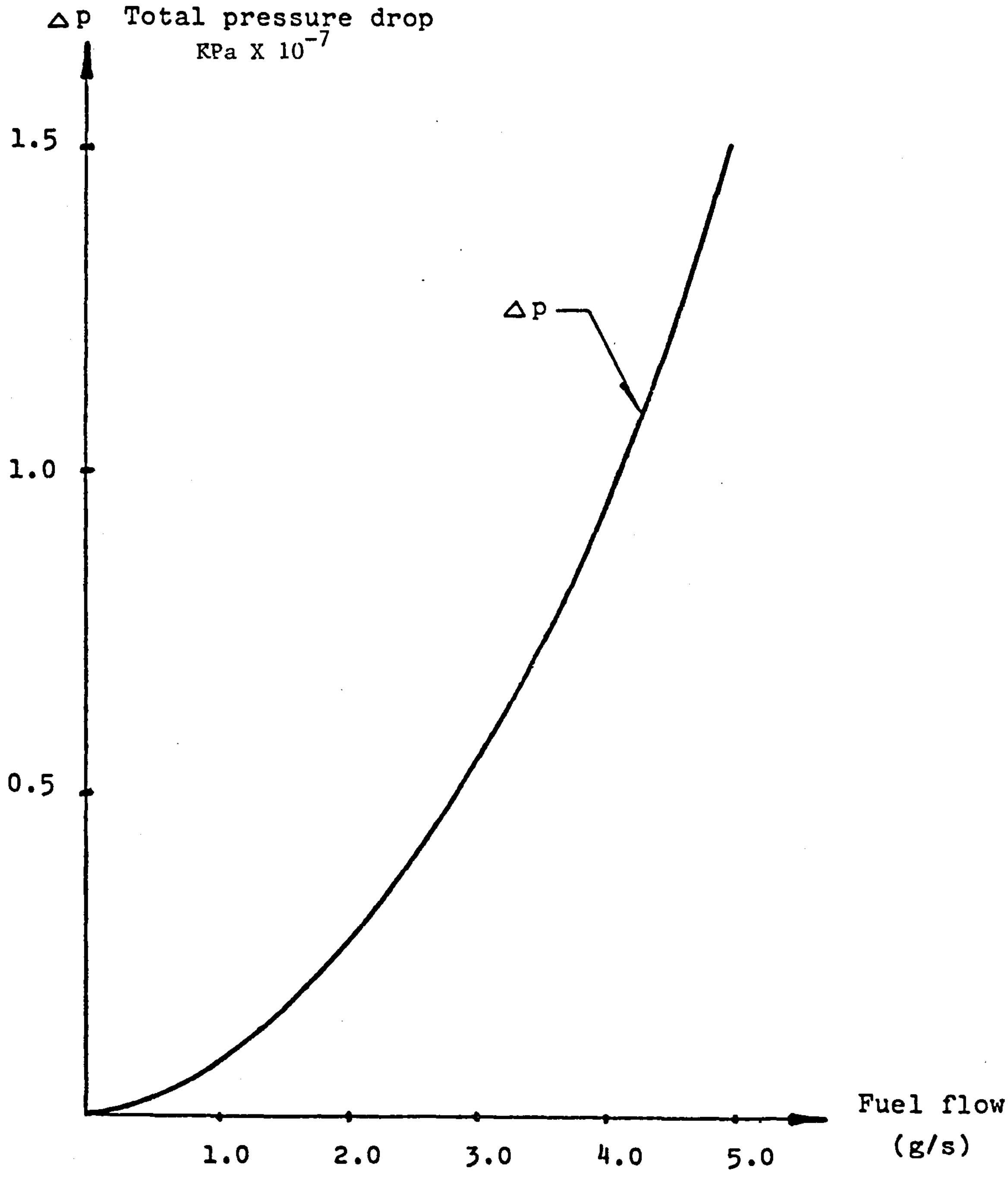


Figure 2.1-3 Total Pressure Drop Versus Fuel Flow

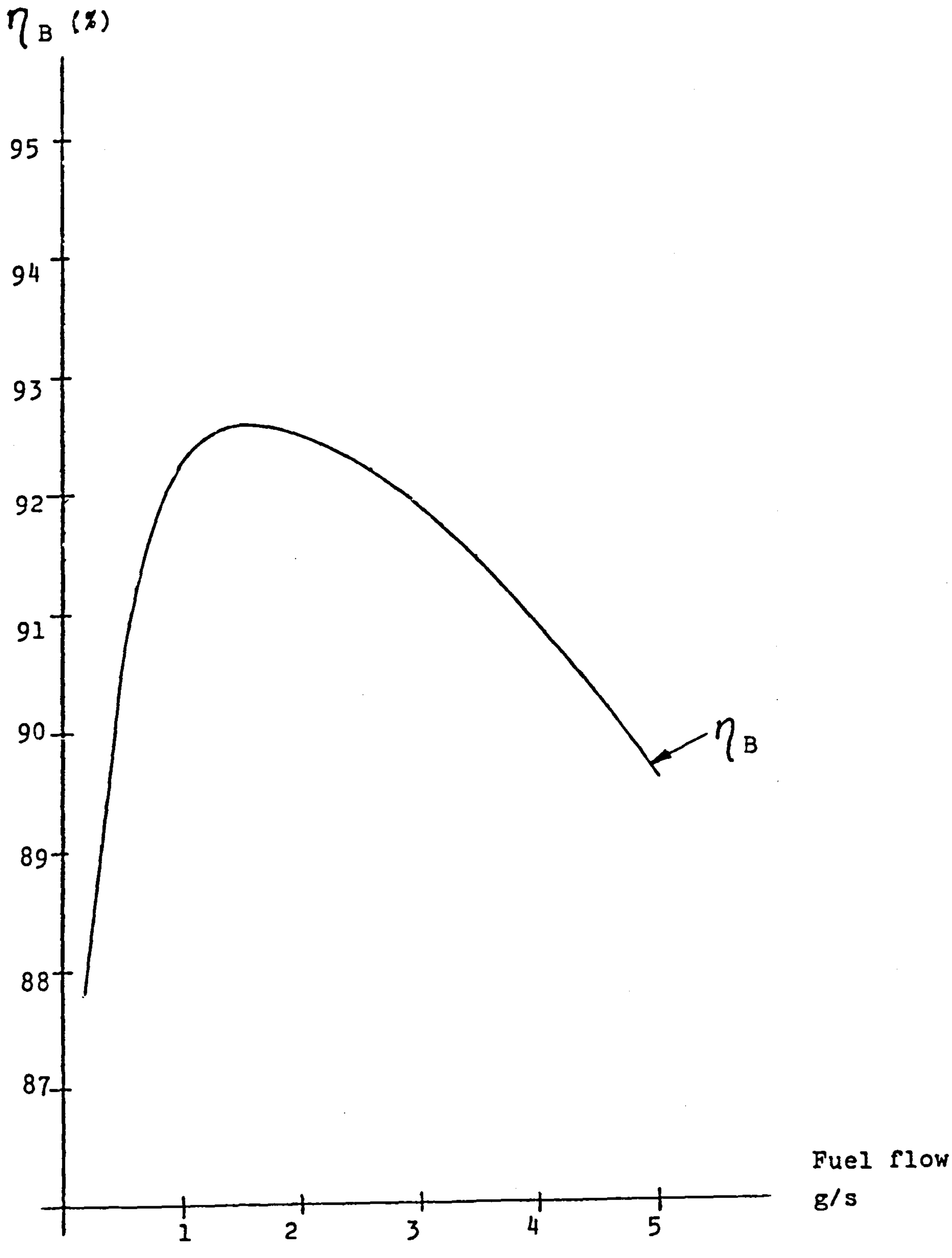


Figure 2.1-4 η_B Versus Fuel Flow

mixture of unburned vaporized fuel and combustion products is forced into the combustion chamber. In the combustion chamber, this mixture is combined with additional air and combustion is completed. To aid in vaporization of the fuel, the inner walls of the vaporization chamber are covered with a course woven net, which prevents larger fuel droplets from entering the combustion chamber without being vaporized.

2.3 Recuperative Air Preheater

The recuperative preheater contains 1100 plates. This compares with the 1200 plates used in the P-40 and Mod I preheater. The dimensions of the RESD preheater are shown in Figure 2.3-1 and a photograph of the P-40 preheater is shown in Figure 2.3-2.

This preheater is a counter-flow type and the matrix is located such that the combustion gas inlet is exactly opposite the second heater tube row. The exhaust is collected in a manifold just below the matrix and from there it is diverted from the engine by means of two exhaust pipes, one on each side of the engine.

The fresh air from the blower is directed into a passage between the outer insulation casing and the outer skin before it enters the preheater matrix. Since the ejectors are located so close to the air outlet, the air is evenly distributed over the entire preheater matrix.

Since both air and exhaust gases enter the matrix in a radial direction and then turn and leave the matrix axially, it was possible, as an alternate design, to make the matrix from a continuously folded strip. This eliminates the costly longitudinal welds at the outer and inner diameters of present matrices used in the P-40 and the Mod I.

Number of Plates		1100
Weight	(kg)	4.25
Total Heat Transfer Area	(m^2)	5.37

Plates

Total Length	(mm)	95
Total Width	(mm)	49.6
Thickness	(mm)	0.1
Crosshead Length Upper	(mm)	21
Crosshead Length Lower	(mm)	12

Corrogated Part of Plates

Bead Angle	(degrees)	15
Width	(mm)	45
Length	(mm)	62
Height	(mm)	0.9
Bead-pitch	(mm)	2.3

Figure 2.3-1 Preheater Dimensions

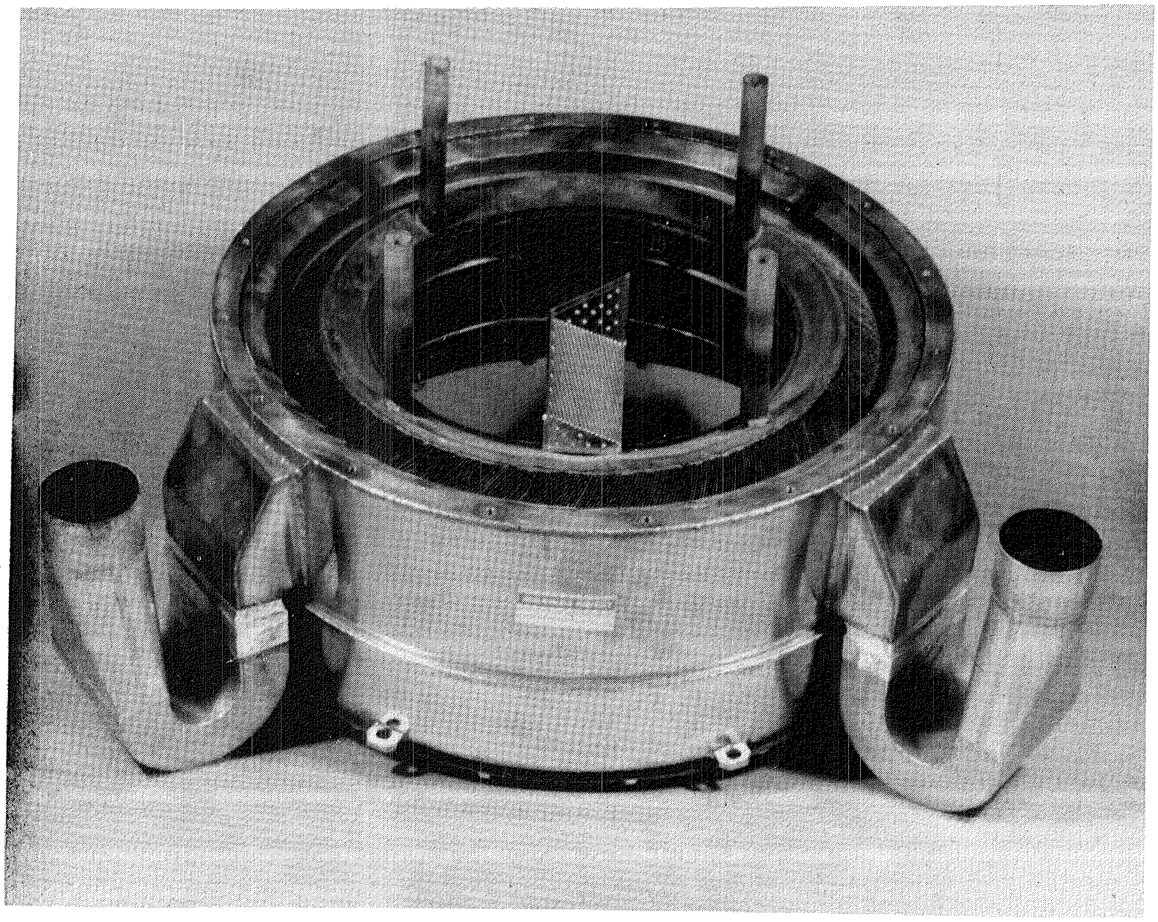


Figure 2.3-2 Preheater With Single Preheater Plate Removed

The matrix is welded to the exhaust manifold and to the two short sheets comprising the ejectors. A compact subassembly without insulation is attached, which is easy to remove for cleaning if clogged by exhaust products.

The size of the preheater housing diameter was reduced because with the air from the blower surrounding the preheater casing, it was possible to decrease the thickness of the insulation while at the same time lowering the outer skin temperature.

On the other hand, this meant that some of the heat leaking through the insulation was picked up by the air which raised the preheater inlet air temperature, especially at part load. This also meant a higher exhaust temperature and a corresponding heat loss. By taking this into account when designing the preheater core, it was possible to keep the outer diameter small without any additional heat loss.

The outer insulation previously mentioned was a vacuum formed, alumina-silica insulation, enclosed in a thin sheet or foil. It was made as a separate part and was easy to install and seal off.

The insulation inside the combustor is made up of a number of alumina-silica parts formed in such a way that they surrounded all of the heater parts and are easily installed. The center parts are anchored by means of a spring loaded bolt, and the outer parts are held in place by the preheater matrix. The preheater is mounted on the cooling water jacket and is attached to the housing by quick-release latches.

The operating life of the preheater matrix is dependent upon the minimum plate temperature realigned during operation; it should be kept above the acid dew point (diesel fuel = 150°C) or the life of the preheater will be drastically reduced. Figure 2.3-3 shows the minimum plate temperature as a function of fuel flow.

The following design factors were used in computing the efficiency of the preheater:

- The air excess factor (λ) was set at 1.1 for all loads.
- The measured percent CGR versus fuel flow, previously shown in Figure 2.1-2, was used.
- The air temperature into the preheater was set at 65°C because its source is through the radiator.
- The heater tube metal temperature was set at 820°C.

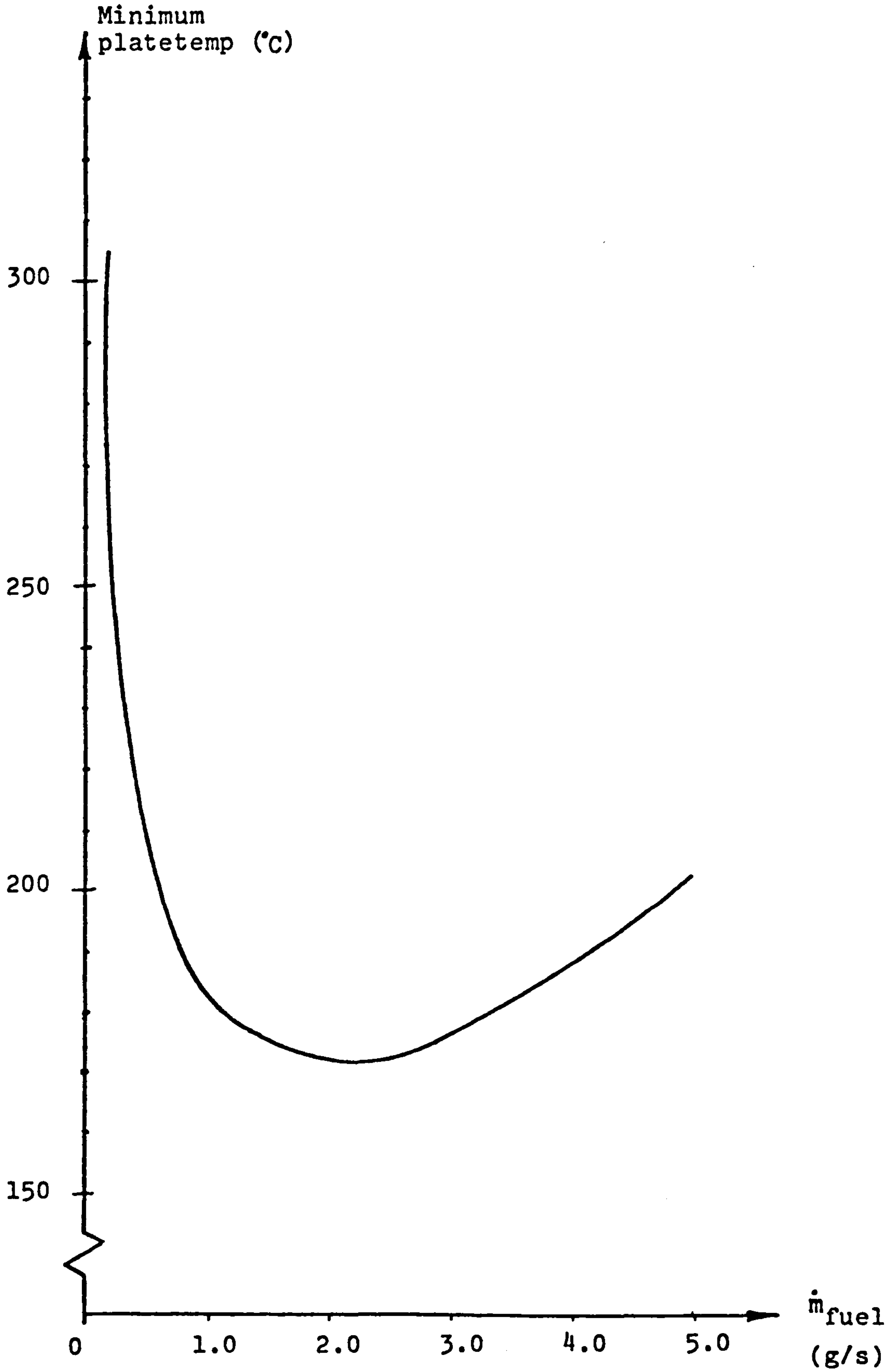
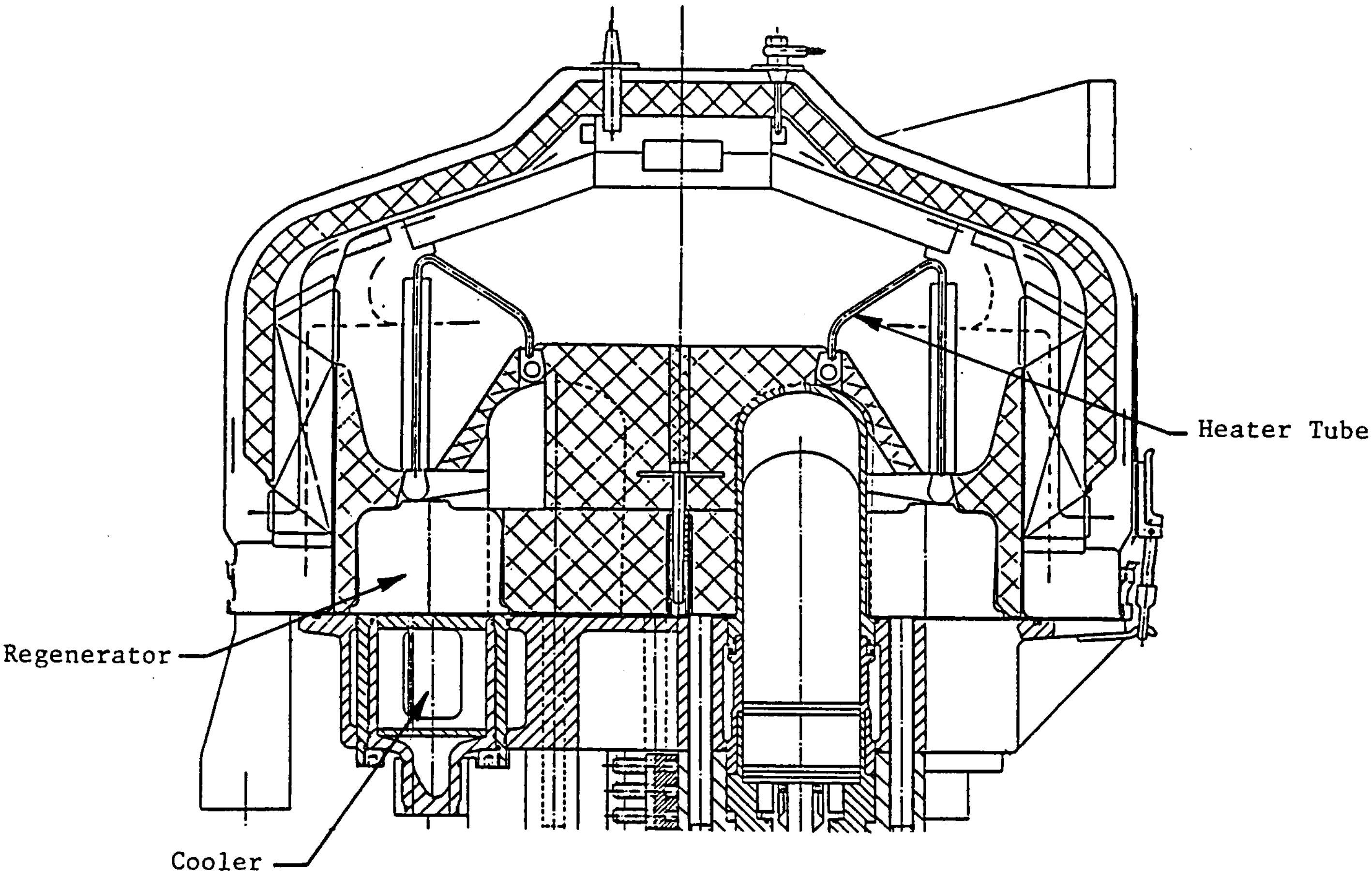
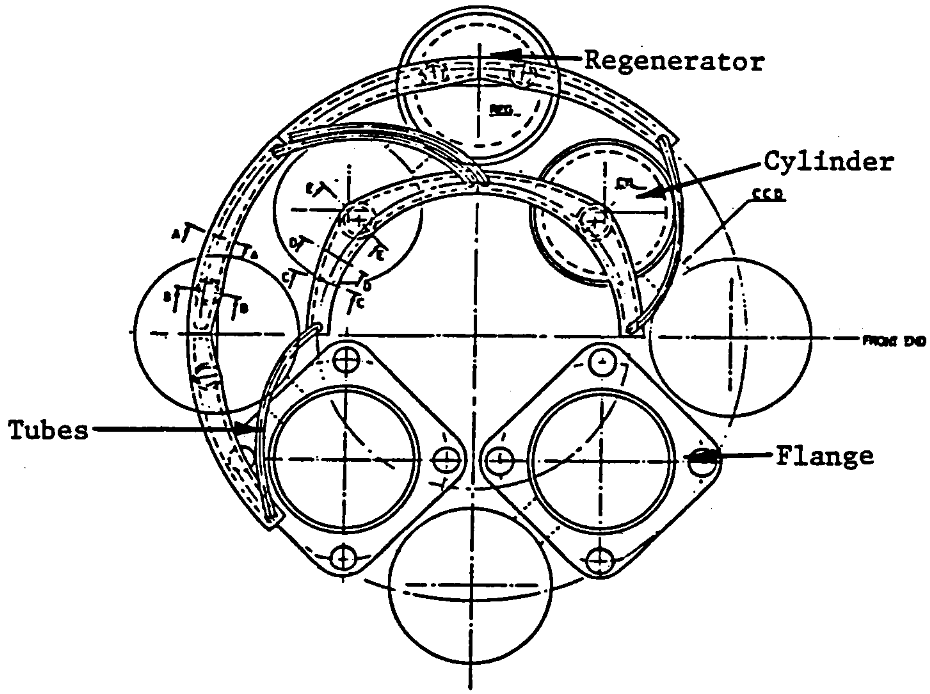


Figure 2.3-3 Minimum Plate Temperature Versus Fuel Flow

SECTION 3.0 HOT ENGINE SYSTEM





Hot Engine System

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3.0 HOT ENGINE SYSTEM

3.1 Heater Head

One regenerator per cylinder, with the housing located as close as possible to the cylinder, was chosen to give a compact design. With involute-shaped heater tubes (45° span), the manifolds are located close to the center of the cylinders and regenerators. Figure 3.1-1 shows a top view layout of two quadrants of the RESD heater head and Figure 3.1-2 shows a photograph of an involute heater head.

Separate manifolds were chosen because integrated manifolds would require more complex castings. (In this study, casting was the only manufacturing method which was considered for cylinders and regenerator housings.) After the manifold castings are machined, the heater tubes are brazed to the manifolds, and these in turn are welded to the machined regenerator and cylinder housings.

Both the cylinder and regenerator housings were designed using a linear creep damaged model, which took into account the actual load spectra over the combined metro-highway driving cycle.

3.1.1 Material

The pressure vessels (cylinder and regenerator housings) were designed for a mean pressure of 15 MPa and a maximum temperature of 800°C (heater tube metal temperature 820°C).

Stress calculations were performed using a finite element program with an assumed temperature gradient in the cylindrical walls. The creep

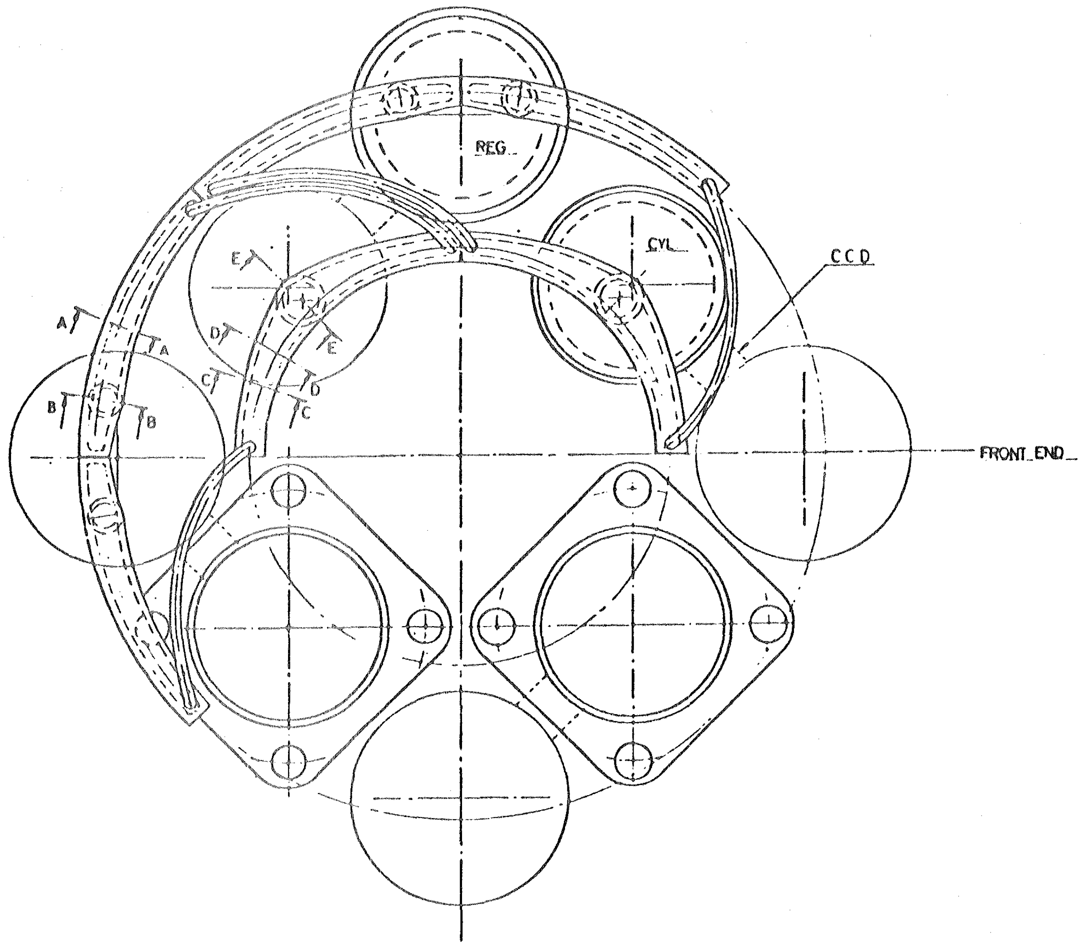


Figure 3.1-1 Heater Head Layout (Top View)

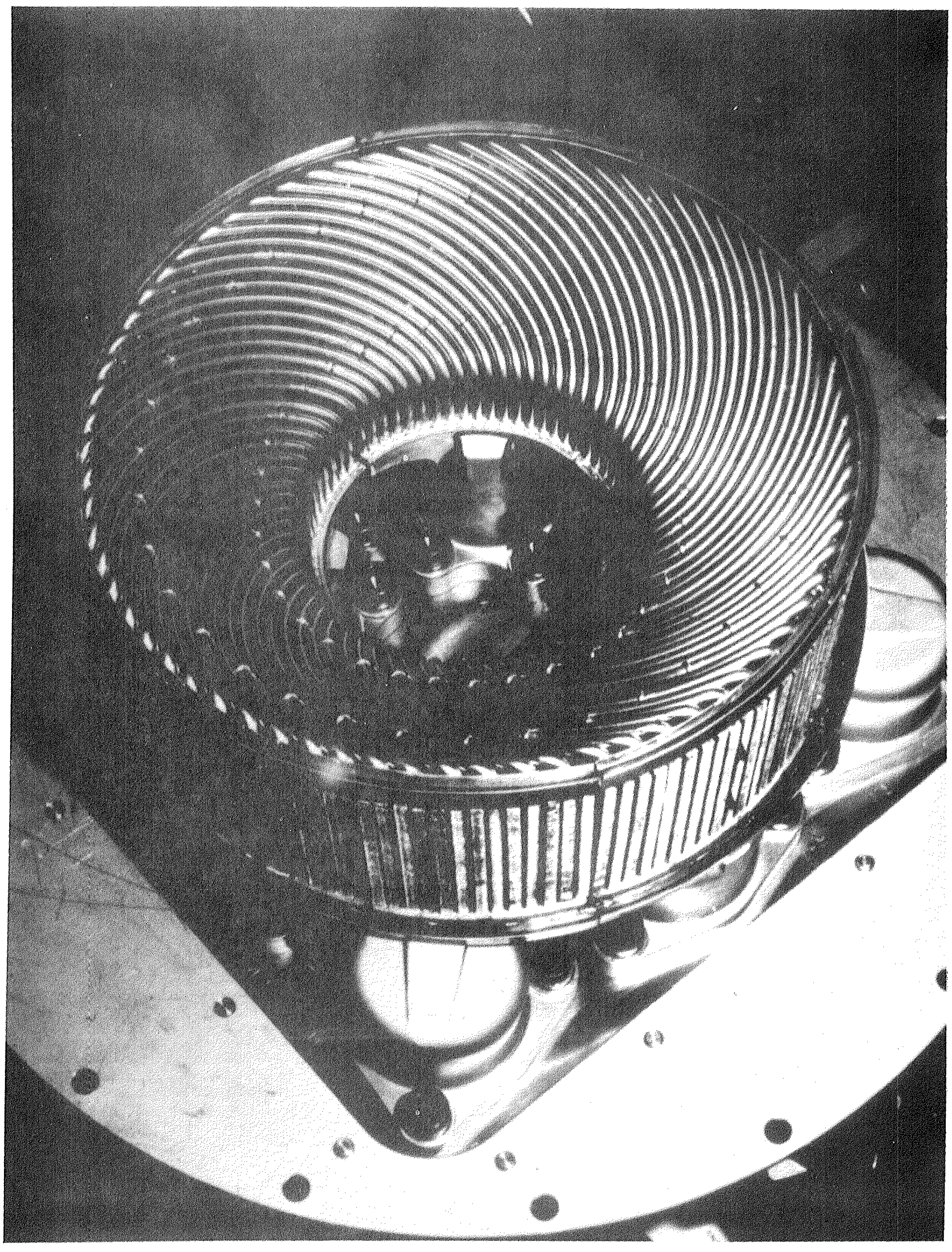


Figure 3.1-2 Involute Heater Head

rupture stress at 800°C and 4000 hours were used to determine the vessel wall thickness. A safety factor of 1.5 was used.

During the design, it was important to keep the pressure vessel housing wall thickness to its minimum to prevent additional heat transfer losses to the cooling water, which would adversely affect output power and result in added cold-start fuel penalties.

The cobalt-base material HS31, which was used in the P-40 and Mod I engines, was not chosen because its properties at 800°C would require much thicker walls in the pressure vessels and would contribute to fuel penalties, as previously discussed. The cost and availability of cobalt-base material also made it undesirable for the automotive application. Climax Molybdenum XF 818 (stainless steel-type material) was chosen because of its characteristics at 800°C working temperatures and its casting properties; XF 818 should also facilitate the brazing/welding assembly procedures. Stress calculation analyses were performed to determine the following thicknesses for cast Climax Molybdenum XF 818:

Cylinder top	4 mm*
Cylinder wall	4 mm - at hot end
Cylinder wall	3.5 mm - at cold end

*Thicknesses were increased locally around the manifold attachments. These values were used to calculate conduction losses and Cold Start Penalty.

Regenerator housing top	8 mm
Regenerator housing wall	7 mm - at hot end
Regenerator housing wall	3 mm - at cold end
Manifold wall thicknesses	3.5 mm

3.2 Regenerator

The regenerator matrix is composed of about 500 to 600 layers of fine mesh screen. Each screen is woven from 50 μ mm (0.00199") stainless steel wire into a 200x200 mesh screen. The layers of screen are stacked randomly into a thin cylindrical casing, compressed, and then sintered to form the regenerator matrix. Figure 3.2-1 is a photograph of the P-40 regenerator.

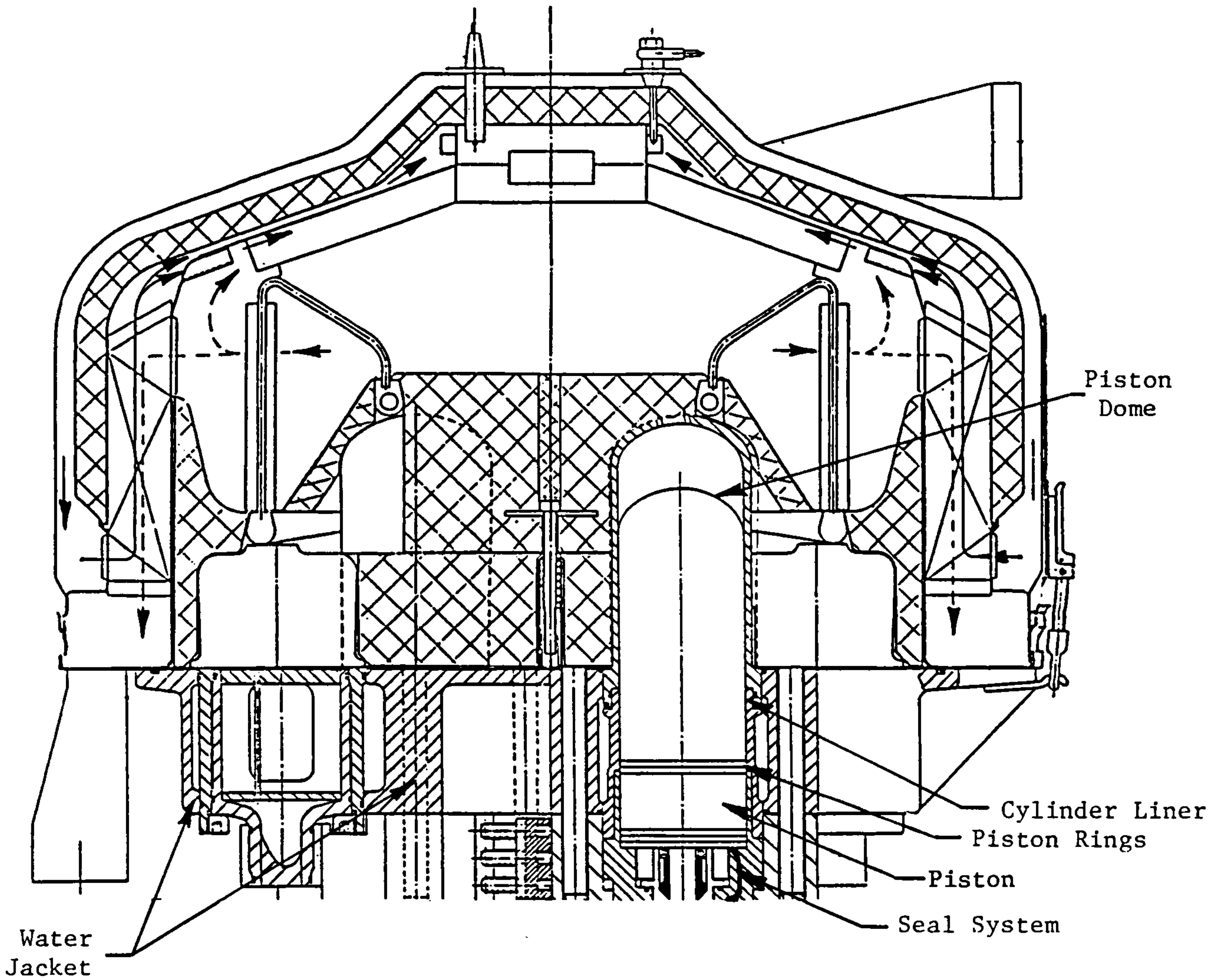
3.3 Cooler

The cooler is made up of small diameter stainless steel tubes, brazed to circular end plates and then to the housing. The tubes may be straight or dimpled and supported midway to prevent buckling. The working gas of the engine, which passes through the hollow tubes, is cooled by the cooling water which circulates around these tubes. The cooling water maintains a temperature of about 50°C. Figure 3.3-1 is a photograph of a cooler.

Figure 3.2-1 Regenerator

Figure 3.3-1 Cooler

SECTION 4.0 COLD ENGINE SYSTEM



Cold Engine System

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4.0 COLD ENGINE SYSTEM

4.1 Water Jacket

The water jacket is the main conduit for supplying cooling water to the engine. The regenerator housing, the cylinder housing, and cylinder liner extend through the water jacket. Mounting bolts for the cylinder housing penetrate the water jacket but do not load the jacket in any manner. The result is a cast, light weight jacket of aluminum with simple water passages. The water jacket top surface extends out to support the air preheater. Figure 4.1-1 shows the water jacket.

4.2 Cylinder Liner-Crosshead Guide

To minimize misalignment and dome gap losses, the cylinder liner and crosshead guide were combined in an integral assembly. The assembly consists of a liner, press fit onto the crosshead guide. Once the liner and guide are assembled, the mating face with the duct plates are machined and the assembly is brazed into the duct plates. The final operation includes machining the cylinder bores and crosshead guide bores.

The entire assembly features an axial groove formed by the press fit of the liner onto the guide which connects the space between the piston rings to the minimum cycle gas pressure line. This connection reduces the leakage caused by ring movement during the reversal of the piston direction. Also featured are three grooves formed at the guide-duct plate joint which act as minimum, maximum, and supply gas pressure lines.

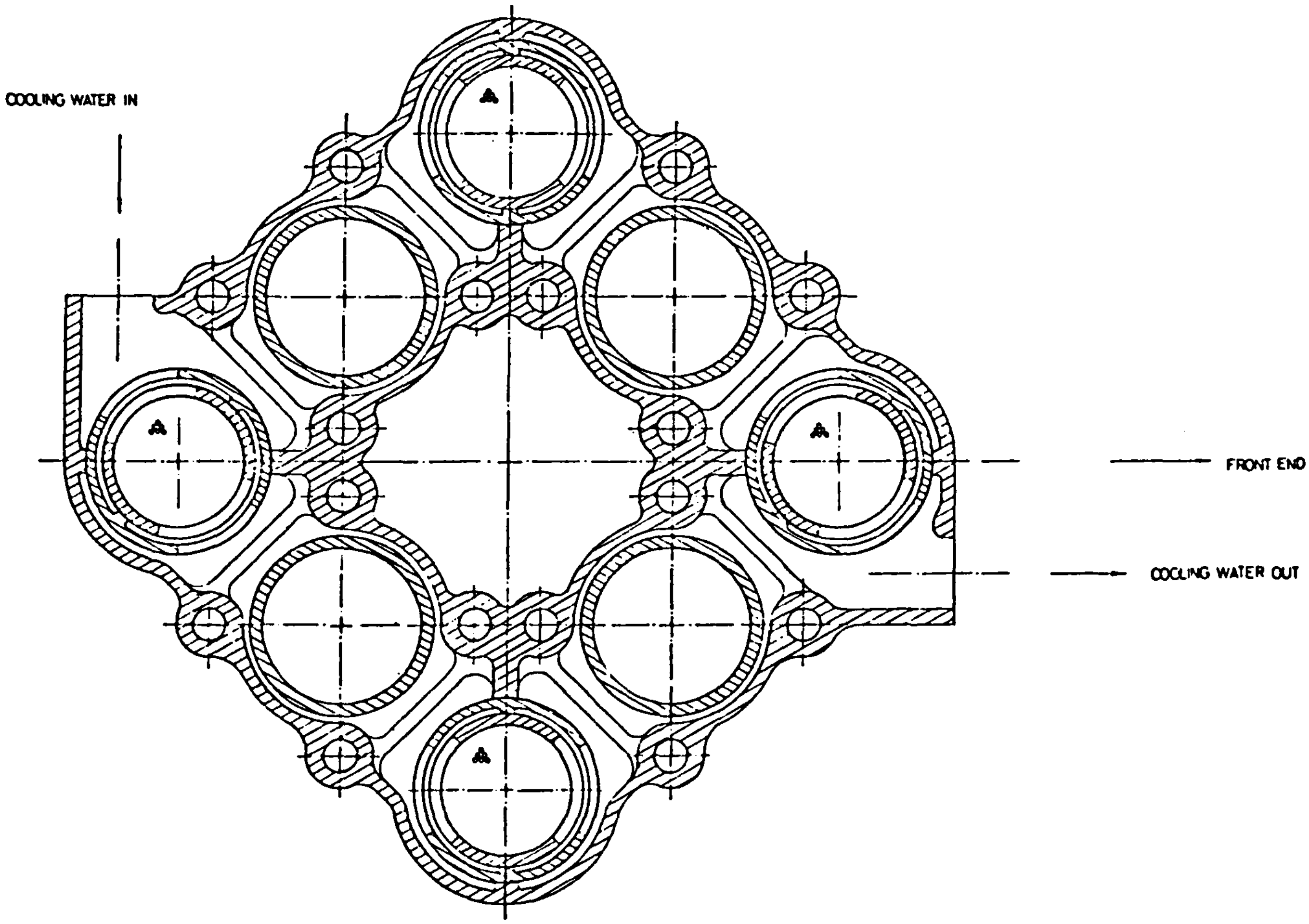

Figure 4.1-1 Water Jacket (Top View)

4.3 Piston and Piston Rings

The piston is made up of two parts: 1) the lower part contains a recess for the rod seal housing and a conical hole for the piston to piston rod alignment; 2) the upper part has a threaded cap for the piston-to-rod attachment and to which the piston dome is brazed or welded. Figure 4.3-1 is a photograph of the piston and piston seal assembly.

The upper part of the piston and the dome is sealed to isolate it from the cycle gas space so that its inner volume does not affect the cycle during changes in pressure. As a result, a smaller hydrogen compressor is used, which minimizes the power consumption of the power control system.

The dome is a thin shell, which contains a thermal radiation shield to keep the threads for the piston rod attachment at the required temperature. The piston is sealed off from the cycle by means of a non-elastomer seal ring between the dome and the piston.

To control leakage across the piston rings during cycle reversal, the minimum cycle pressure was introduced between the rings to prevent the rings from moving from one side of the piston ring grooves to the other. Figure 4.3-2 shows the gas line assembly.

4.4 Piston Rod Seal

The piston rod seal design chosen was a spring loaded, Pumping Leningrader (PL) main seal. Contrary to the P-40 and Mod I, there was no cap seal in this design because tests performed with PL seals running both with and - without cap seals indicated that the PL seal performed equally well without

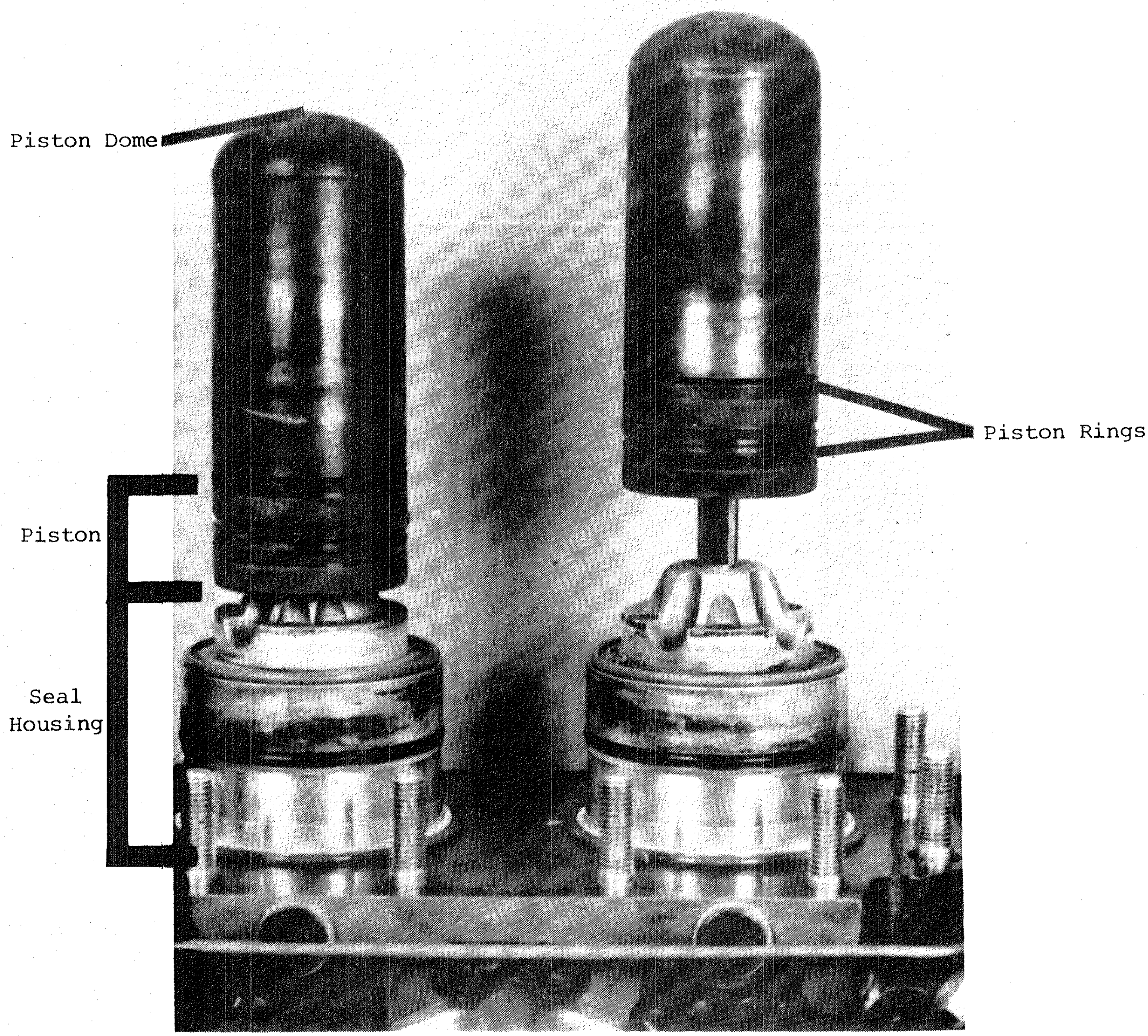

Figure 4.3-1 Piston Seal Assembly

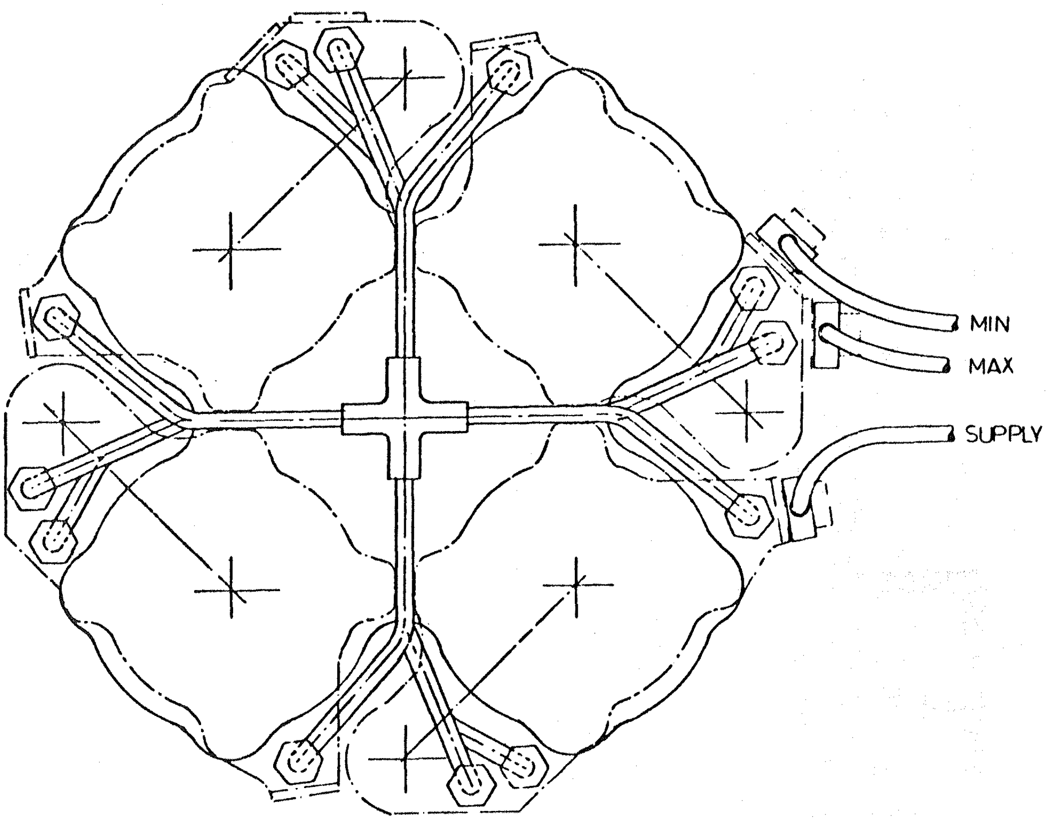

Figure 4.3- 2 Gas Line Assembly (Viewed from bottom of cold engine system)

the cap seal. The elimination of the cap seal removed a part that experience had shown to be extremely susceptible to wear; this change will increase the seal assembly's reliability.

All the seal elements located in the seal housing are made up of two parts: the lower part with the seat for the Leningrader seal, and the cover. This design allows the two parts to be locked together with a pin in such a way that the seal parts could be removed or assembled as one unit. Figure 4.4-1 is a photograph of the seal housing cartridges for the P-40.

When installing the seal elements, it is important that radial misalignment between the piston rod and the seals is minimized; otherwise, excessive wear could occur. To prevent such a misalignment, the seal housing floated and is guided by the piston rod, which in turn is held in alignment by the crosshead and the piston.

4.5 Duct Plate

The duct plate serves numerous functions. It contains several ducts or passages which eliminate much of the external plumbing required on previous engines. The duct plate contains provisions for:

- Passage between the compression space and the cooler;
- Supply gas duct to the piston rod seal housing;
- The maximum and minimum cycle gas pressure ducts;

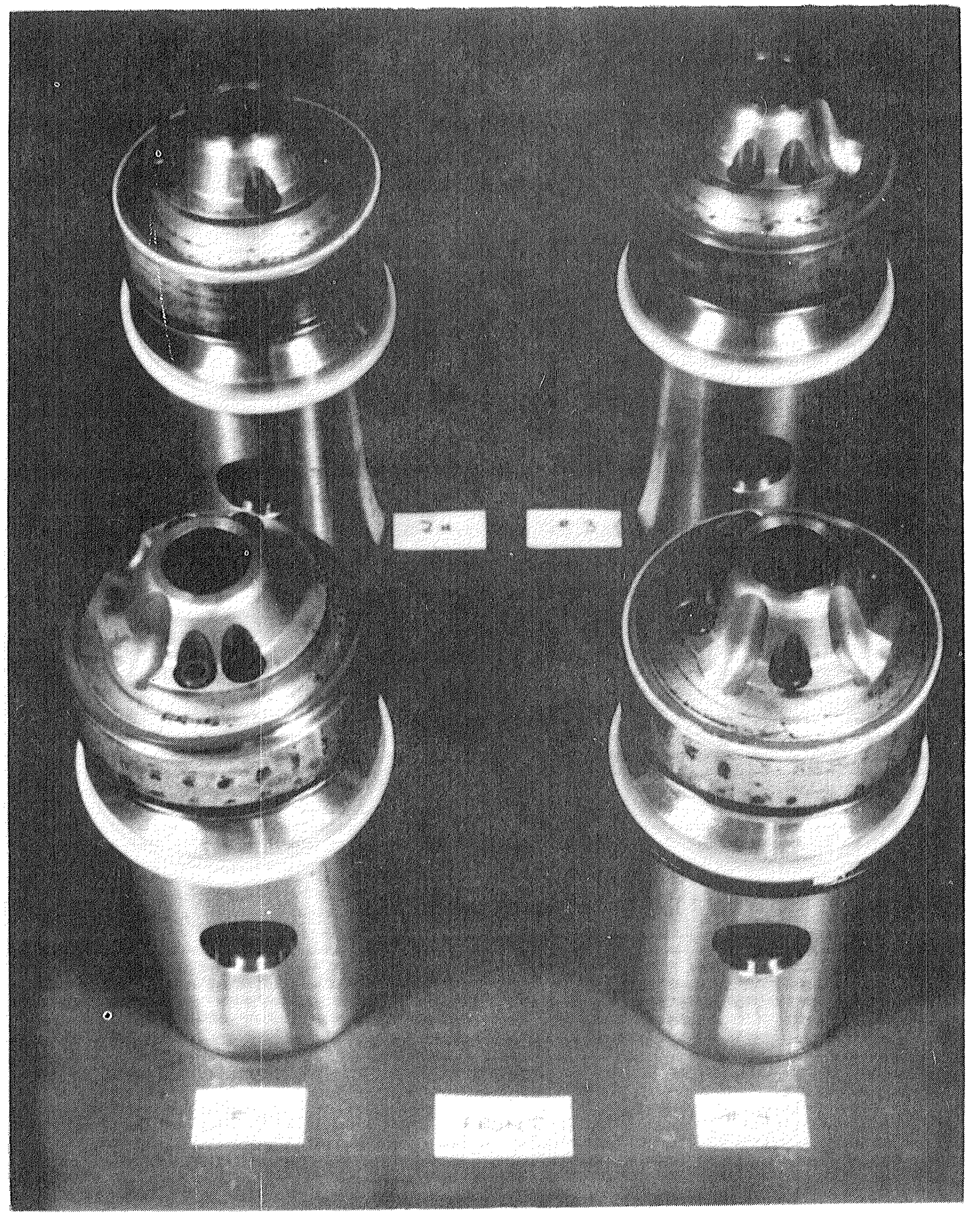

Figure 4.4-1 Seal Housing Cartridges

- The passages required for maintaining the minimum cycle pressure between the piston rings;
- Housings for all check valves.

All connections between the cycles and the gas lines from the power control valve were designed to be of equal length and symmetrically arranged. Figure 4.5-1 shows the duct plate assembly.

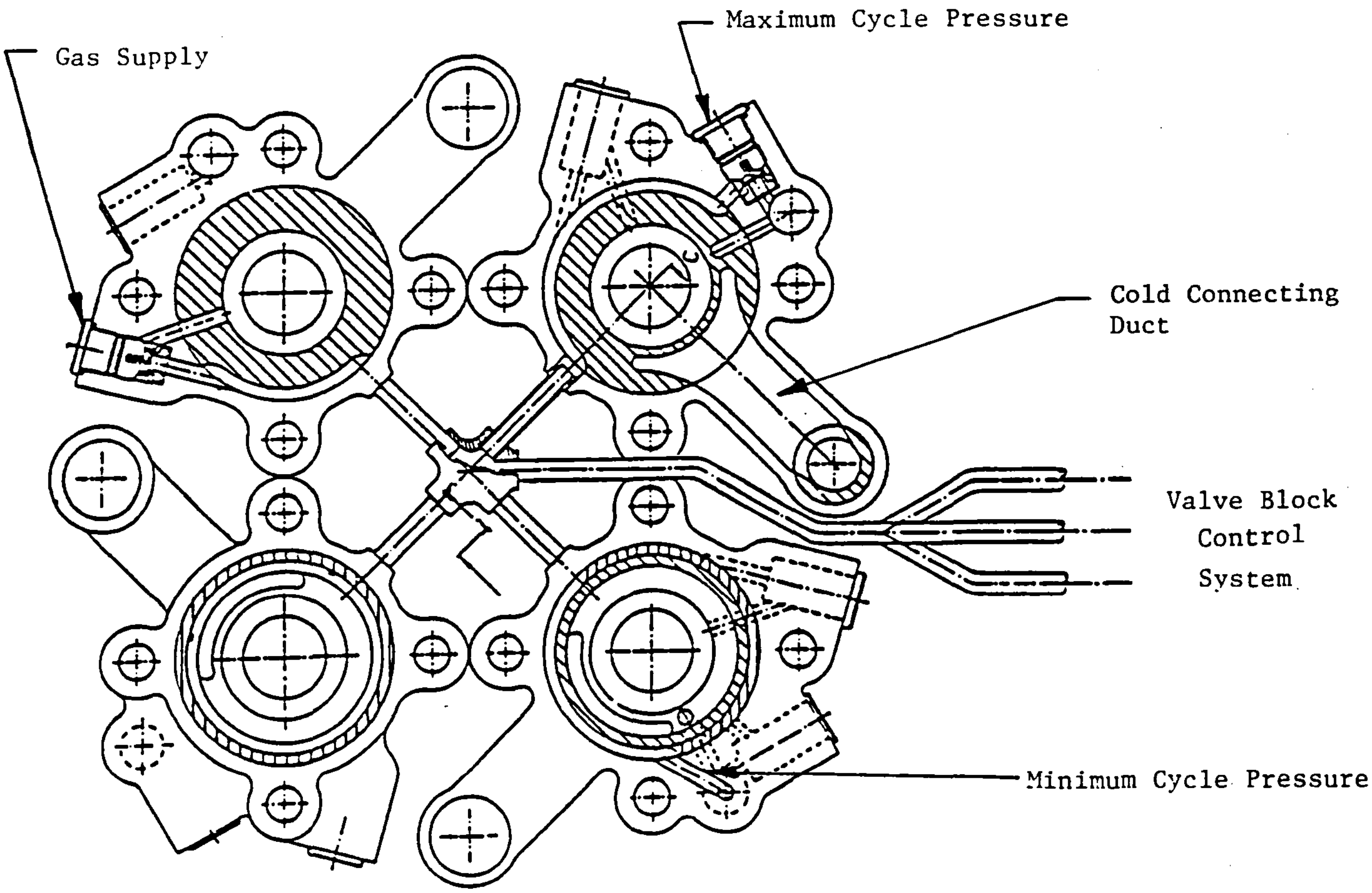

Figure 4.5-1 Duct Plate Assembly (Viewed from Bottom of Cold Engine System)

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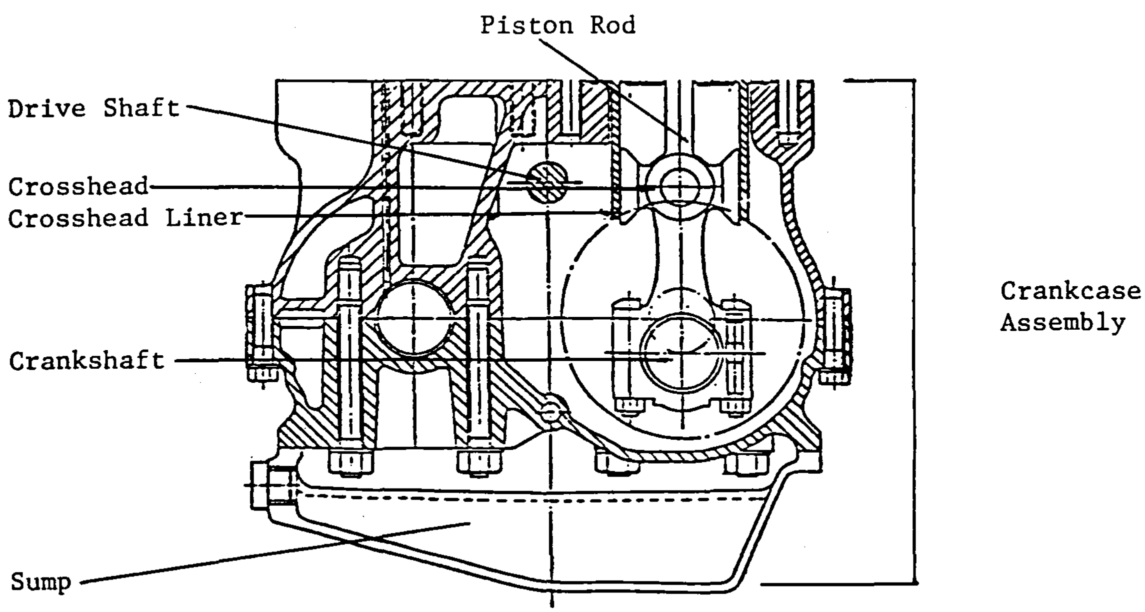
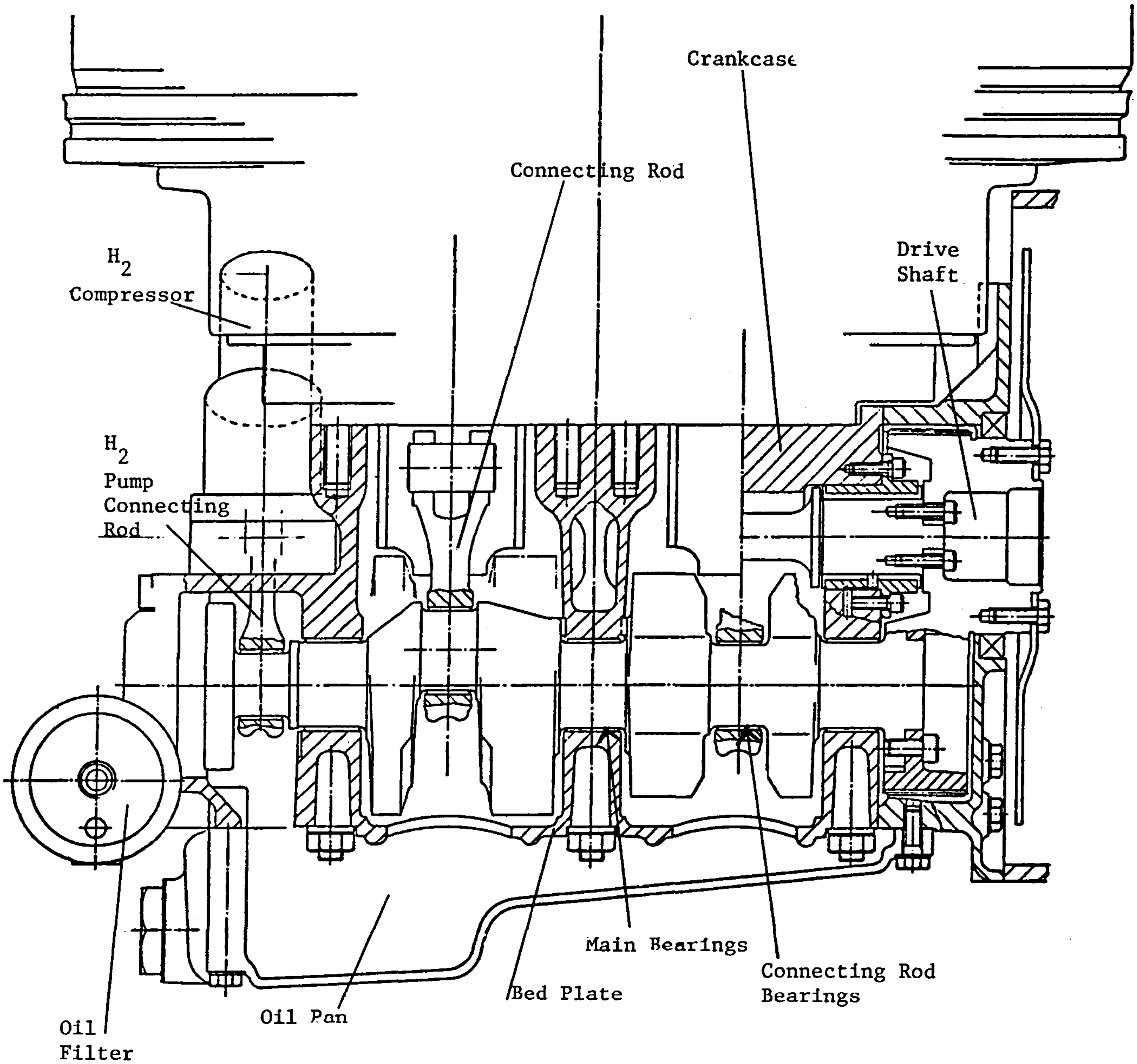
SECTION 5.0 ENGINE DRIVE SYSTEM



Engine Drive System (End View)



Engine Drive System (Side View)

5.0 ENGINE DRIVE SYSTEM

The assembly was designed for low weight, low cost, small package size, and ease of manufacture. The low weight requirement necessitated that aluminum be used in all major castings and that all unnecessary spare volume be removed.

The selected crankcase/bedplate arrangement gives a very rigid box construction. Pre-assembled units of the oil pump/filter water pump, and intercasing/gears and crankshafts allow for fast production assembly.

The crankshafts and connecting rods were designed as modular iron castings, which resulted in minimum machining of pins, journal and balance weights.

To gain maximum rigidity and strength for a U-4 type engine, a crankcase and bedplate construction was preferred to the simple, low skirt crankcase with bearing caps. This allowed the loads applied by the cylinder head studs to be dissipated through the side walls of the crankcase into the bedplate and main bearing studs.

Figure 5.0-1 shows a crankcase assembly; Figure 5.0-2 shows a parallel crankshafts and bed plate assembly; Figure 5.0-3 shows a rod assembly.

The lubrication oil system is somewhat different from that of the baseline engine in that the pump is fitted to the crankcase assembly and is driven from the crankshaft, which allows for a low profile sump to be used.

The water pump is also directly fitted to the crankcase and the overall length is reduced by using a single twin-row angular contact bearing to support the impeller shaft.

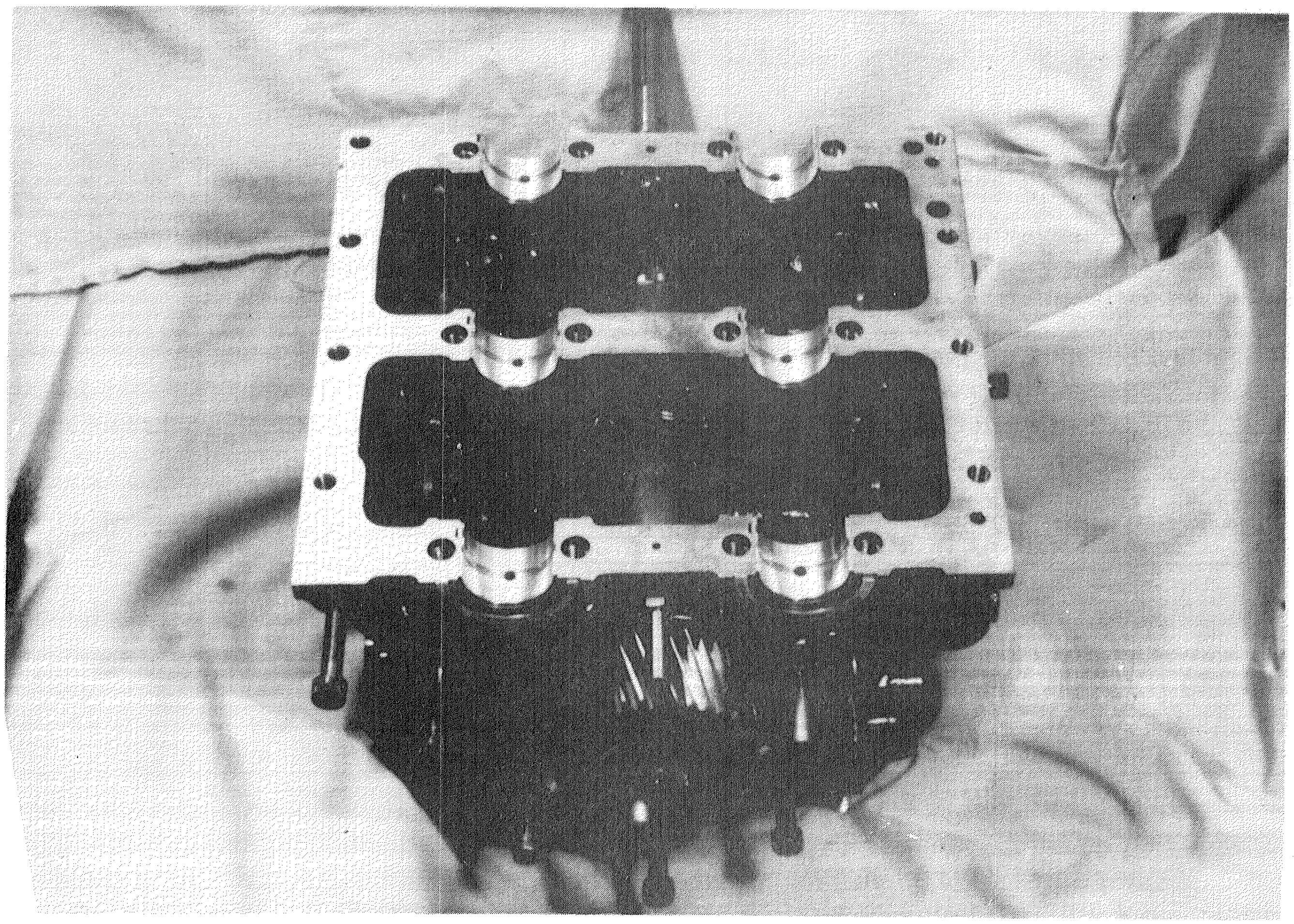


Figure 5.0-1 Main Crankcase Assembly

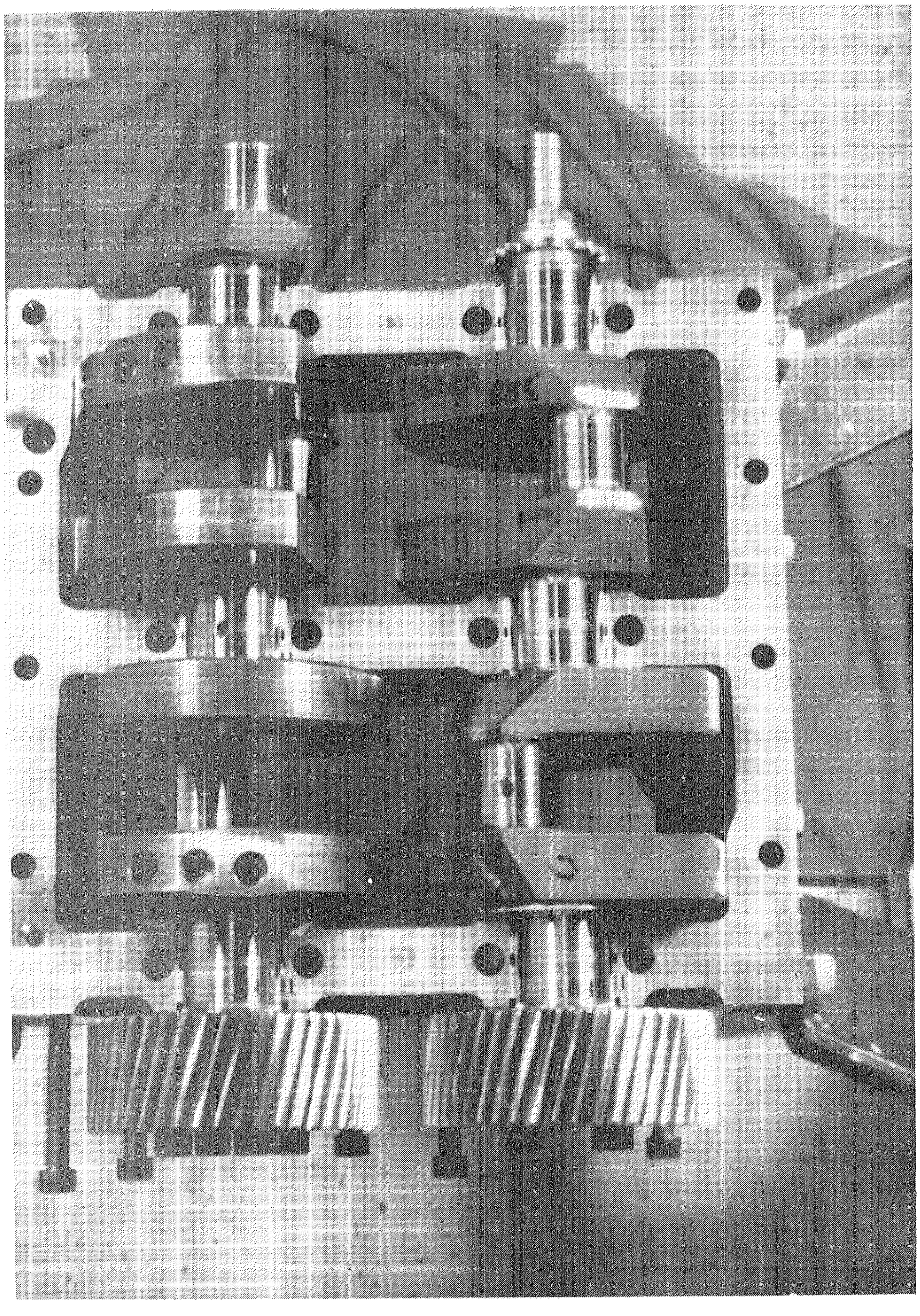


Figure 5.0-2 Parallel Crankshafts With Bedplate Assembly

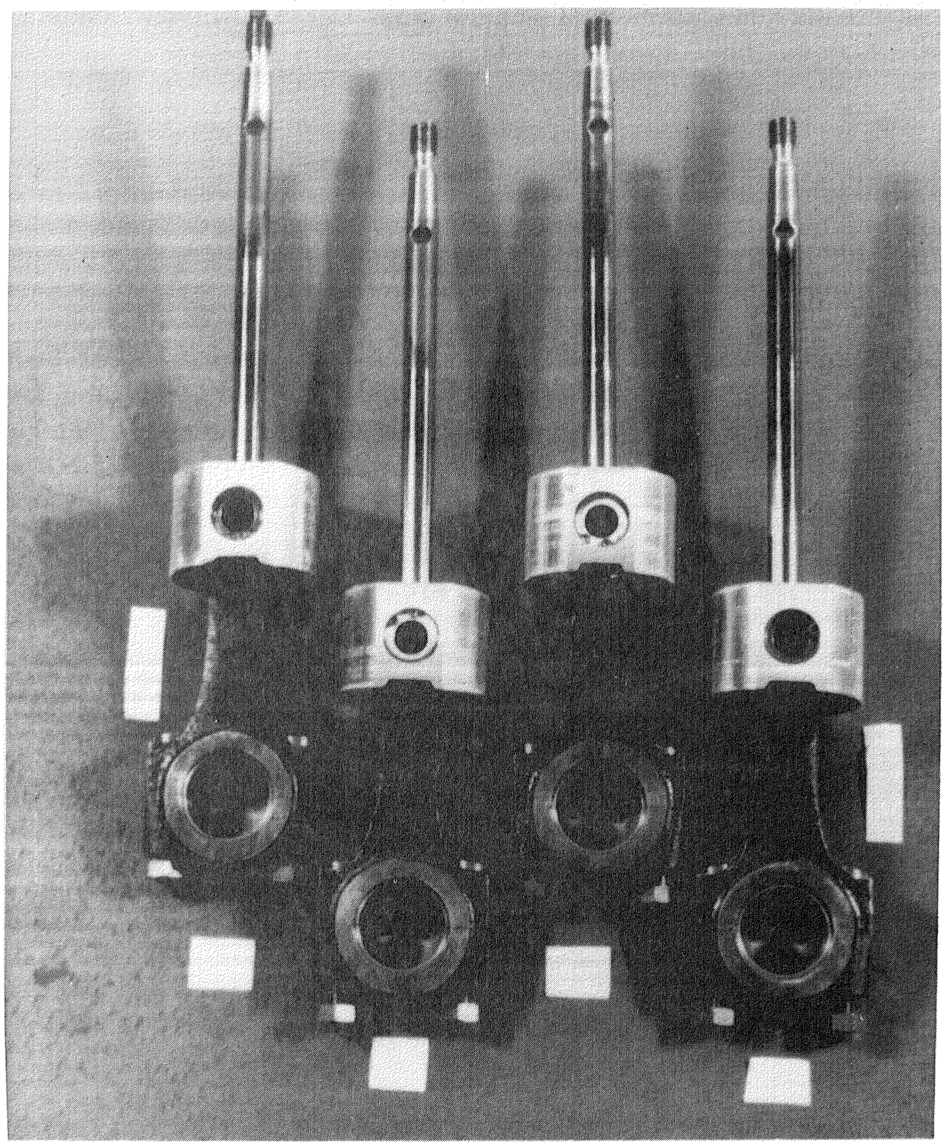


Figure 5.0-3 Piston and Connecting Rod Assembly

The hydrogen compressor is also directly mounted and is driven by the other crankshaft.

Figure 5.0-4 shows the underside of the crankcase assembly and the lubrication system for the P-40.

The drive system was designed for an all-gear synchronization with balance weights located on both sides of the connecting rod bearings. The crankshaft main bearing diameters are 38 mm (compared to 45 mm for the Mod I); the connecting rod bearings are 35 mm (compared to 42 mm for the Mod I); the height of the crankcase is 125 mm from the crankcase upper face to the crankshaft centers (compared to 130 mm for the Mod I).

To provide better matching of the RESD with available torque converters/ transmissions, an engine speed increaser was needed. This change in output shaft speed was achieved by changing the gear ratio to 1.4 from the two crankshafts to the output shaft.

The engine balance shaft must rotate at engine speed to overcome the reciprocating pitch couple, but due to the speed difference with the step-up drive, it was necessary to provide an external shaft to accomplish the required balancing. In addition to the added balancing shaft, the output shaft was split at the front main bearing bulkhead, allowing it to be driven by gears from one of the crankshafts so that the variator would be driven at engine speed, optimizing its performance and durability.

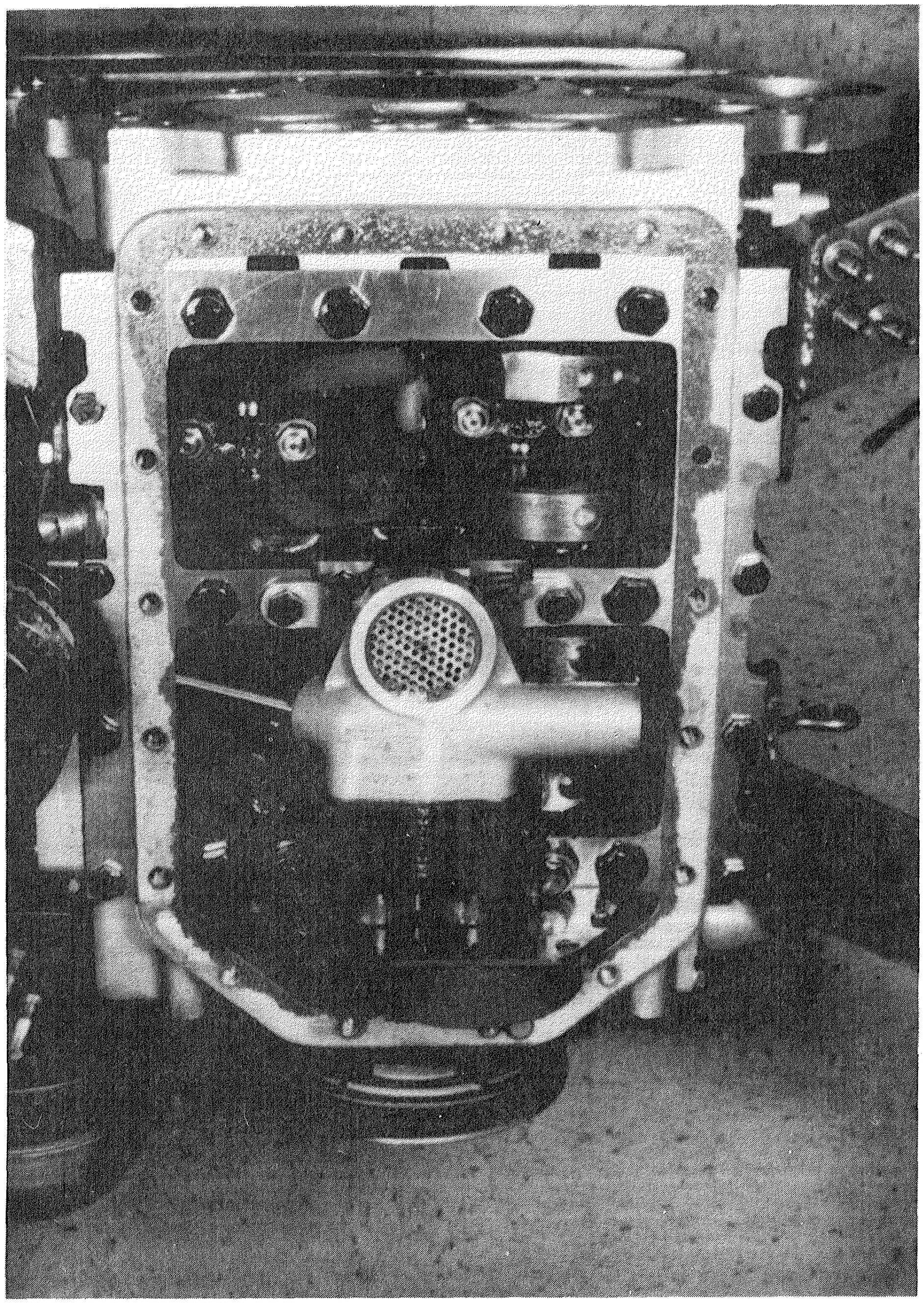


Figure 5.0-4 Bottom View of Crankcase Assembly

SECTION 6.0 POWER CONTROL SYSTEM AND AUXILIARIES

6.0 POWER CONTROL SYSTEM AND AUXILIARIES

6.1 Power Control System Selection

The power control systems developed by Philips, Ford, and USSw (United Stirling of Sweden) are combinations of these basic methods:

1. Mean pressure variation.
2. Pressure amplitude variation due to external dead volume.
3. Pressure amplitude variation due to variation in stroke length.
4. Pressure amplitude variation due to gas temperature variation.
5. Phase control due to bypass.
6. Phase control due to crankshaft angle variation.

These real systems were investigated as the power control system for the Reference Engine System (RES). Table 6.1-1 shows the suitability of these systems as RES power control systems.

Some Real Power Control Systems

			Basic System
A.	USSw	Mean Pressure Control System	1 + 5
B.	USSw	Dead Volume Control System	2+1+5
C.	USSw	Turk System	1+(5)
D.	Philips	Variable Swash-Plate Control System	3 + 5
E.	Ford	Hybrid System (P_{mean}-Dead Volume)	1+2+5

Power Control System	Major Negative Performance Characteristic	Reference Engine Action Required	Remarks
USSw Mean Pressure Control	No	Yes	
USSw Dead Volume Control	Yes	No	Power output in steps. Bad idle fuel consumption.
USSw Turk System	Yes	No	Weight problem. Long supply time
Variable Swash-* Plate Control	?	No	Needs another drive concept.
Ford Hybrid System	No	Yes	

*Note: The variable swash-plate was not considered as a RES power control system, because the RES drive concept was determined to be a dual crankshaft unit.

Table 6.1-1 Possible Power Control Systems for the RESD

From this initial system selection, two systems remain:

A. United Stirling — Mean Pressure Control System

B. Ford — Hybrid System

Computer calculations were made to evaluate the performance of the Ford Hybrid System versus the United Stirling mean pressure control system. These calculations did not show any significant difference in efficiency for either system. But because the Ford Hybrid System would require additional components and would contain more hydrogen, it was not selected as the RES power control system.

Thus, the United Stirling mean pressure control system was selected for the RES.

6.2 The RES Power Control System - Specification and System Description

The major parts of the mean pressure control systems are the hydrogen storage tank, the control valve block, and the hydrogen compressor.

- System Specification:

    1) The system should not contain more than 80 grams of hydrogen.

    2) Response specifications for the RES when a change in power is applied from idling to full power at constant speed:

        a) The lag time at the drive shaft should not be more than 0.15 seconds after a change of load.

        b) The torque at the drive shaft must reach 90% of maximum available torque within 0.6 seconds (See Figure 6.2-1).

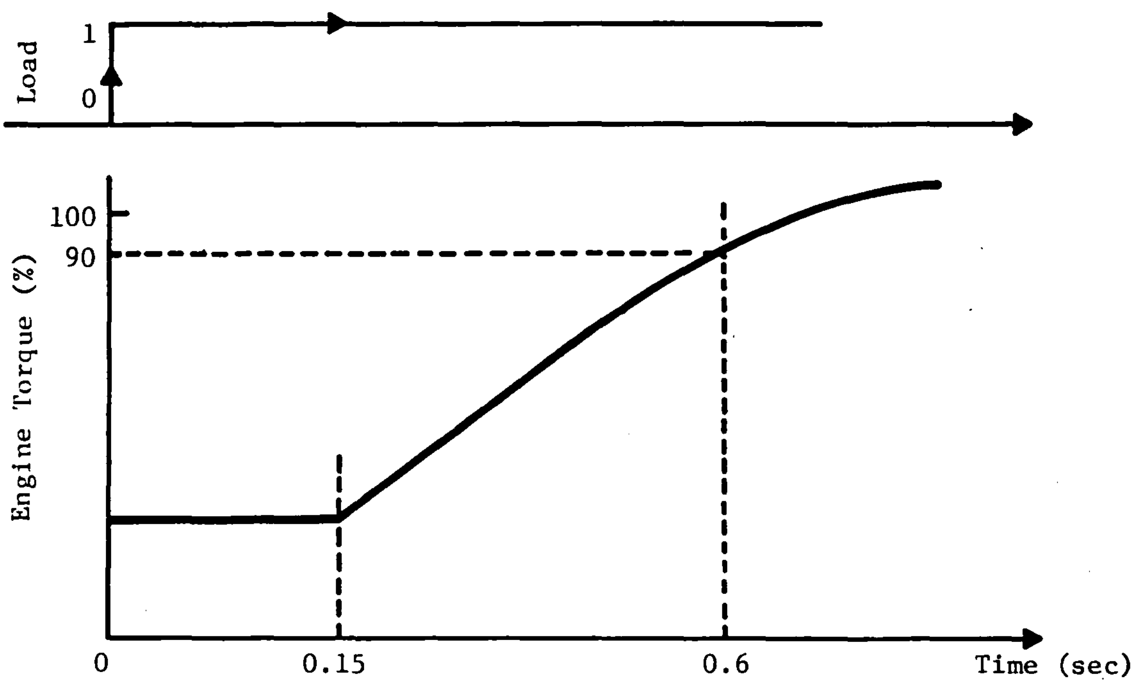


Figure 6.2-1 Engine Torque Versus Time

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c) During power decrease (engine braking), the torque at the driveshaft must decrease to 20% of its maximum value within 0.2 sec. (See Figure 6.2-2).

3) No hydraulic servo oil system.

4) Reliability comparable to other external components.

5) Hydrogen storage tank must be of sufficient capacity to maintain a minumum of a six month recharge schedule.

- System Description

  - The Hydrogen Storage Tank

    The storage tank is a 4 liter tank with a maximum charge of 230 MPa. (See Figure 6.2-3). Table 6.2-1 lists the specifications for the hydrogen storage tank.

  - Control Valve Block

    The electrically actuated control valve consists of an integrated modular block design, which contains most of the active parts in the control system. All valves and transducers were specifically designed for this application in order to reduce production costs, complexity, and to facilitate maintenance.

  - Hydrogen Compressor

    The hydrogen compressor is a single stage, double-acting pump. The pumping performance of the compressor should

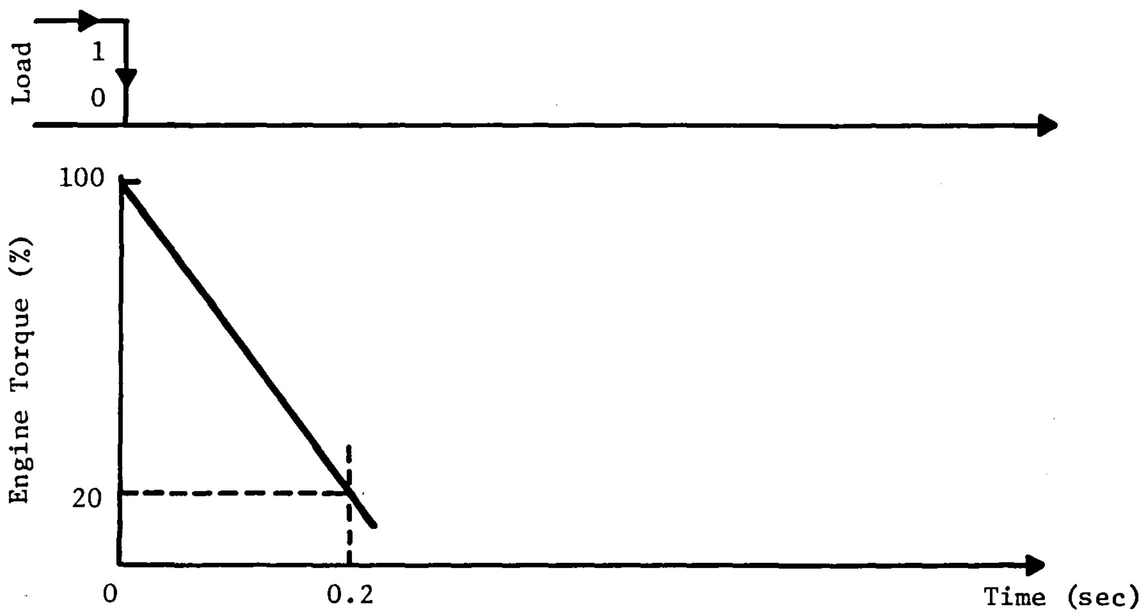

Figure 6.2-2 Engine Torque Versus Time

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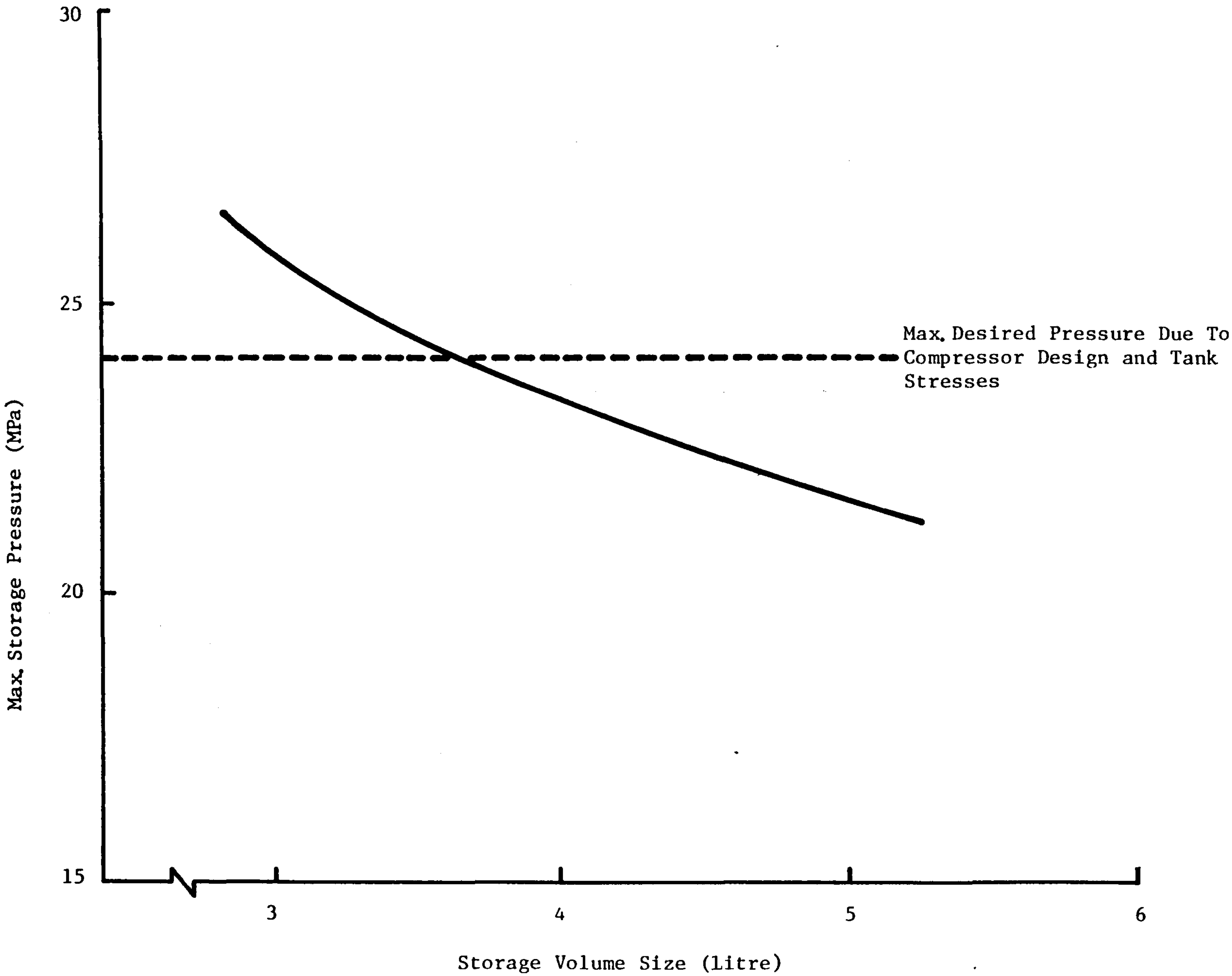

Figure 6.2-3 Maximum Charge Pressure of the Storage Tank as a Function of Tank Size for Reference Engine

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	Tank Size (Liter) 4
Amount of hydrogen (kg) in storage tank (70°C) at full engine power. Lowest tank pressure 15 MPa.	0.0417
Amount of hydrogen (kg) in storage tank at idling (70°C)	0.0497
Tank pressure (MPa)	17.9
Reserve capacity due to leakage (210 normal liters)	5.3
Total max charging pressure MPa	23.2

Table 6.2-1 Specifications for Hydrogen Storage Tank

follow the curve shown in Figure 6.2-4. See Section 6.10 for the power consumption of the compressor during short circuiting.

The control system electronics is a computerized microprocessor.

6.3 Combustion Air Blower

The calculations were based on a combustor without a bypass valve to reduce power demand.

$$\text{Blower power } p_B = \left(\frac{\Delta p \; \dot{m}_a}{\rho_a}\right)\left(\frac{1}{\eta_B \, \eta_v}\right)$$

Δp = total air pressure drop (Pa) (Figure 6.3-1)

$\dot{m}_a$ = Air mass flow (kg/s) (Figure 6.3-2)

ρ_a = Air density 1.19 kg/m^3

η_B = Blower efficiency (Figure 6.3-3)

η_v = Variable ratio belt drive efficiency (Figure 6.3-3)

6.4 Water Pump

The power demand for the water pump is shown in Figure 6.4-1.

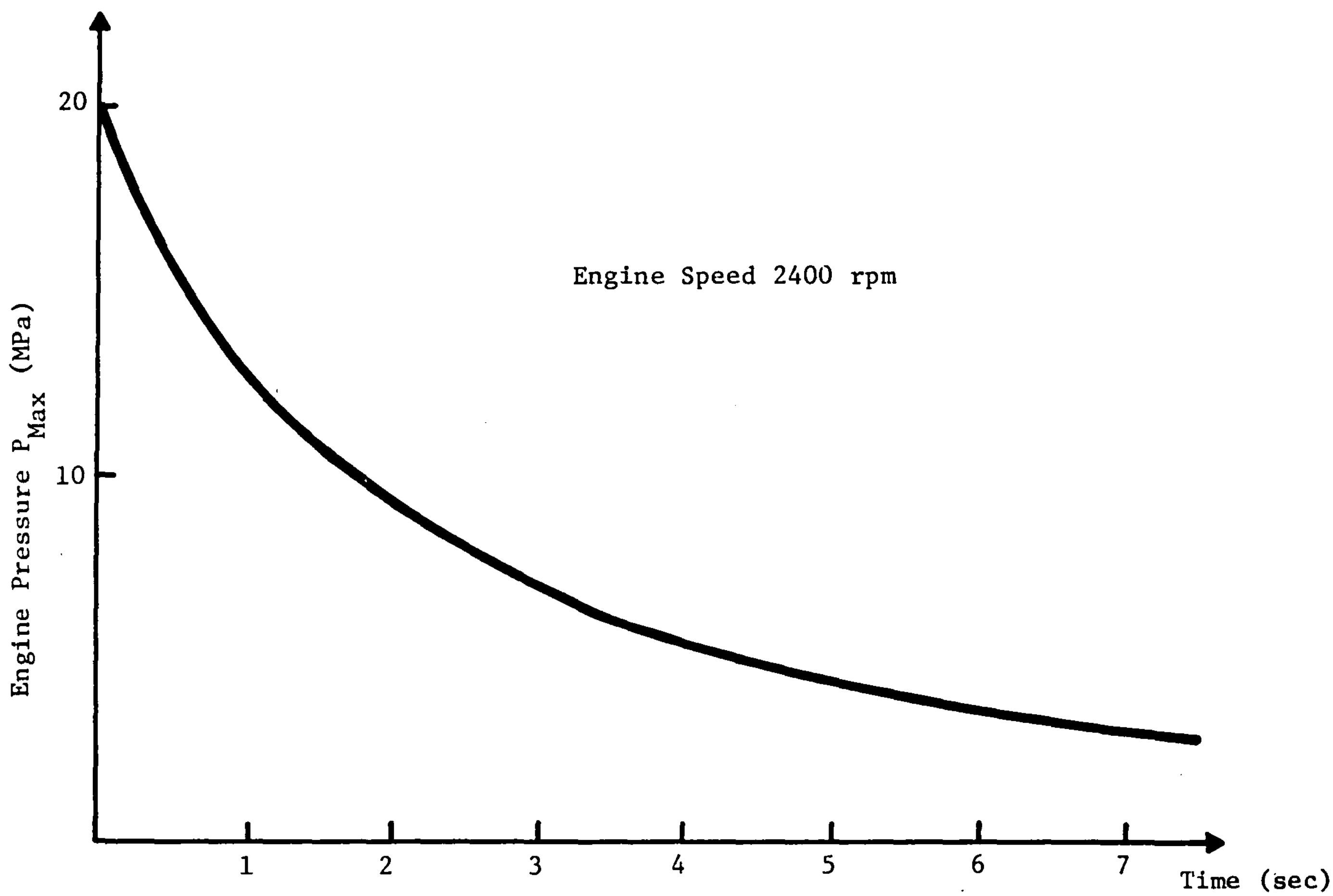

Figure 6.2-4 Pumping Performance of Compressor

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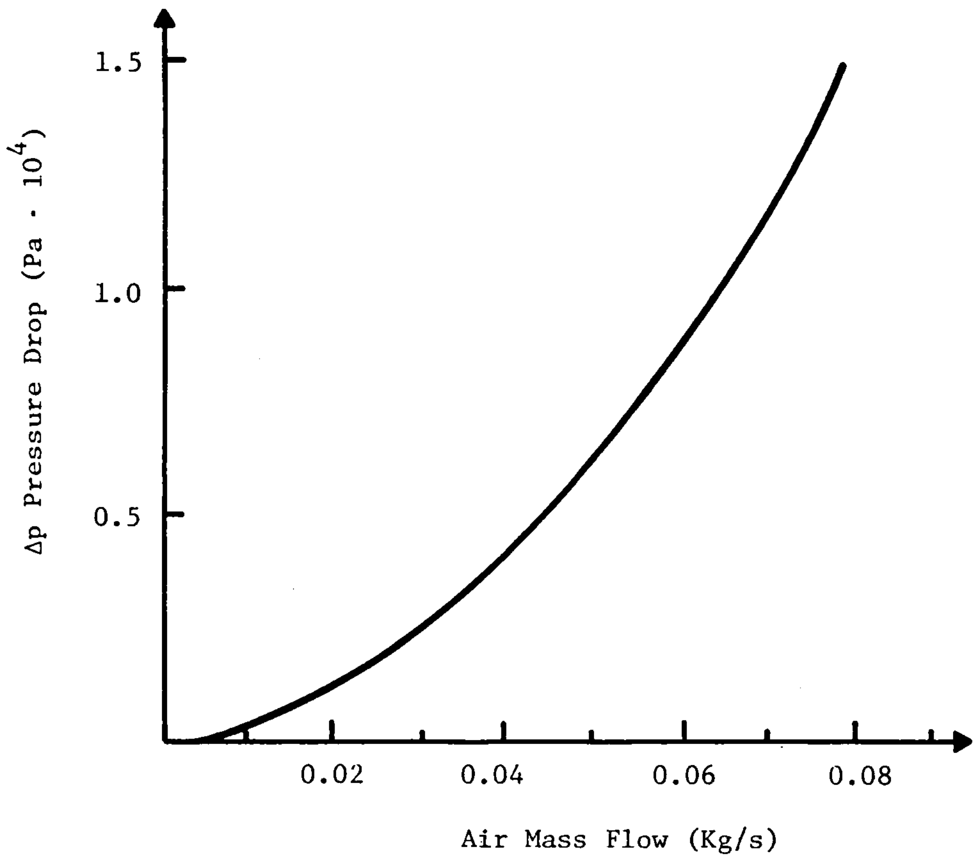

Figure 6.3-1 Total Engine Pressure Drop

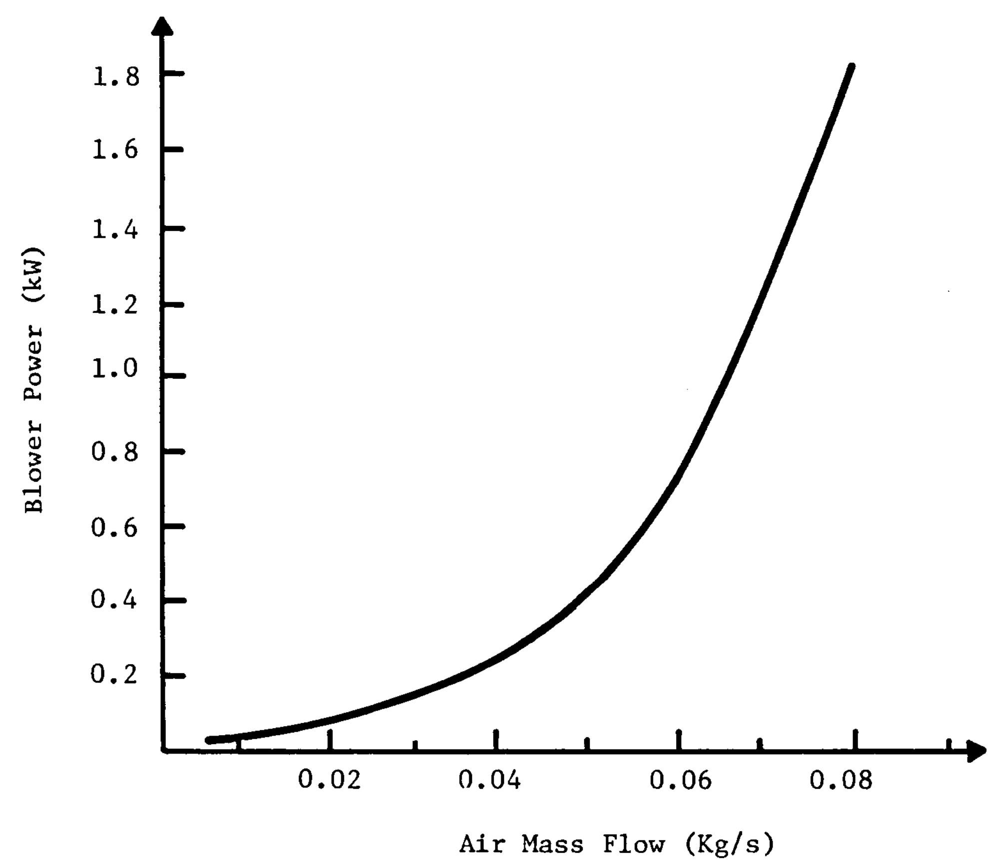

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Figure 6.3-2 Blower Power Demand

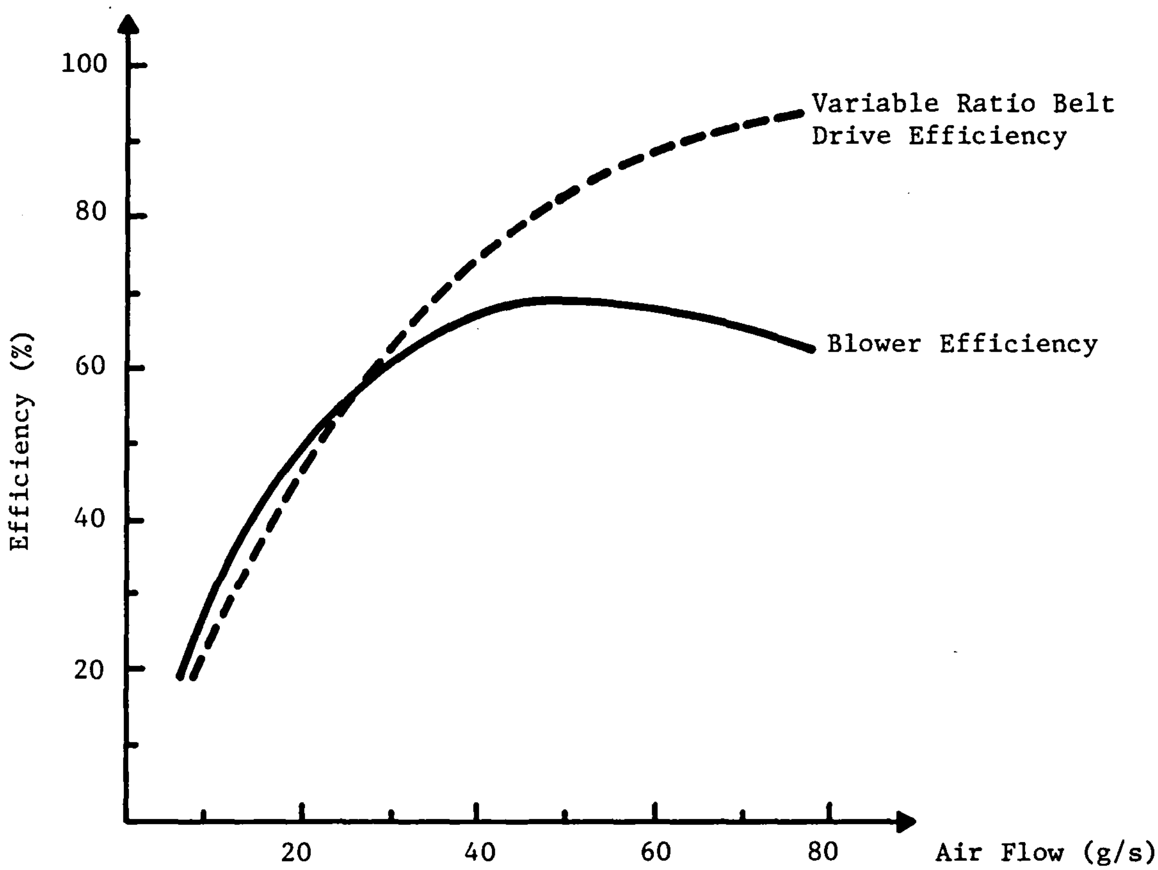

Figure 6.3-3 Blower and Belt Drive Efficiency

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6.5 Alternator

The alternator power demand was based on the following electric devices:

Electronics	0.024 kW
Ignition	0.015 kW
Fuel Pump	0.009 kW
Solenoid Valves	0.030 kW
Fuel Valve	0.006 kW
Air Throttle	0.006 kW
TOTAL	0.090 kW

In addition to the 0.090 kW continuous power demand, there are intermittent power demands from the electric blower motor, starting motor, lights, and radio of approximately 0.055 kW. However, this power does not affect the mileage calculations if a fully charged battery is assumed. The alternator power demand is shown in Figure 6.5-1.

6.6 Lubricating Oil Pump

The power consumption curve for the oil pump is shown in Figure 6.6-1.

6.7 Power Steering Pump

The power consumption curve for the power steering pump is shown in Figure 6.7-1.

6.8 Hydrogen Gas Compressor

The power consumption curve for the hydrogen gas compressor is shown in Figure 6.8-1.

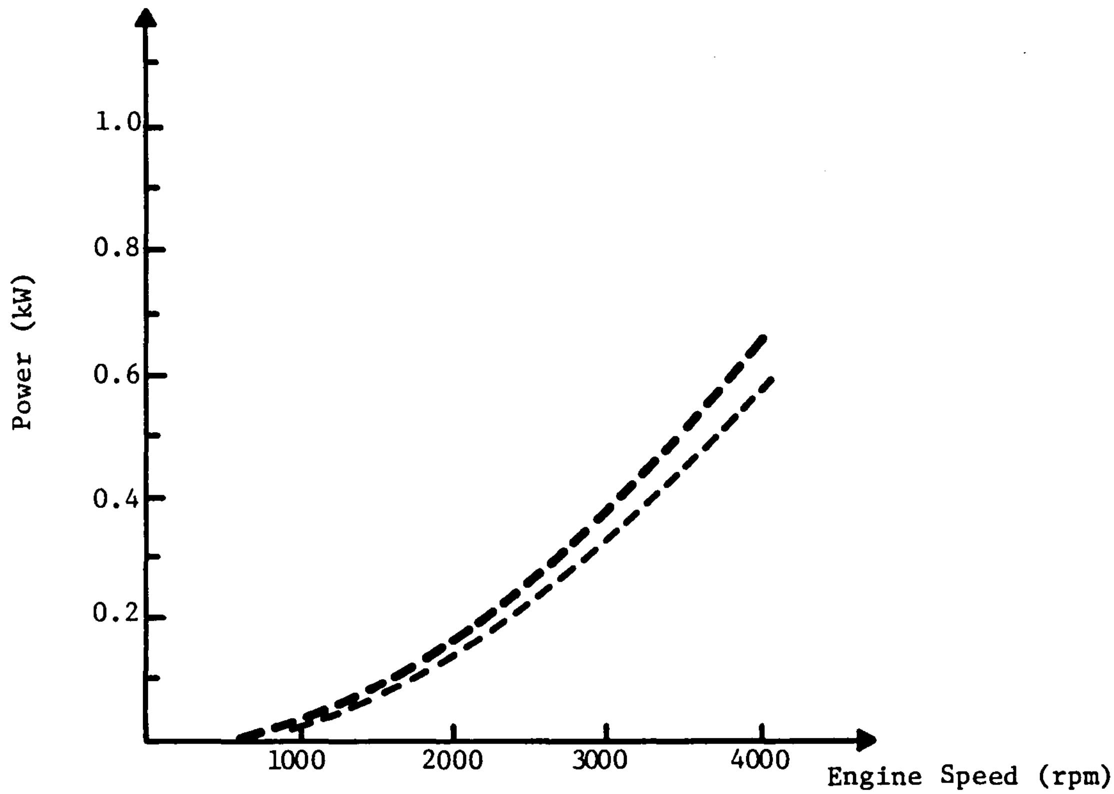

Figure 6.4-1 Water Pump Power Demand

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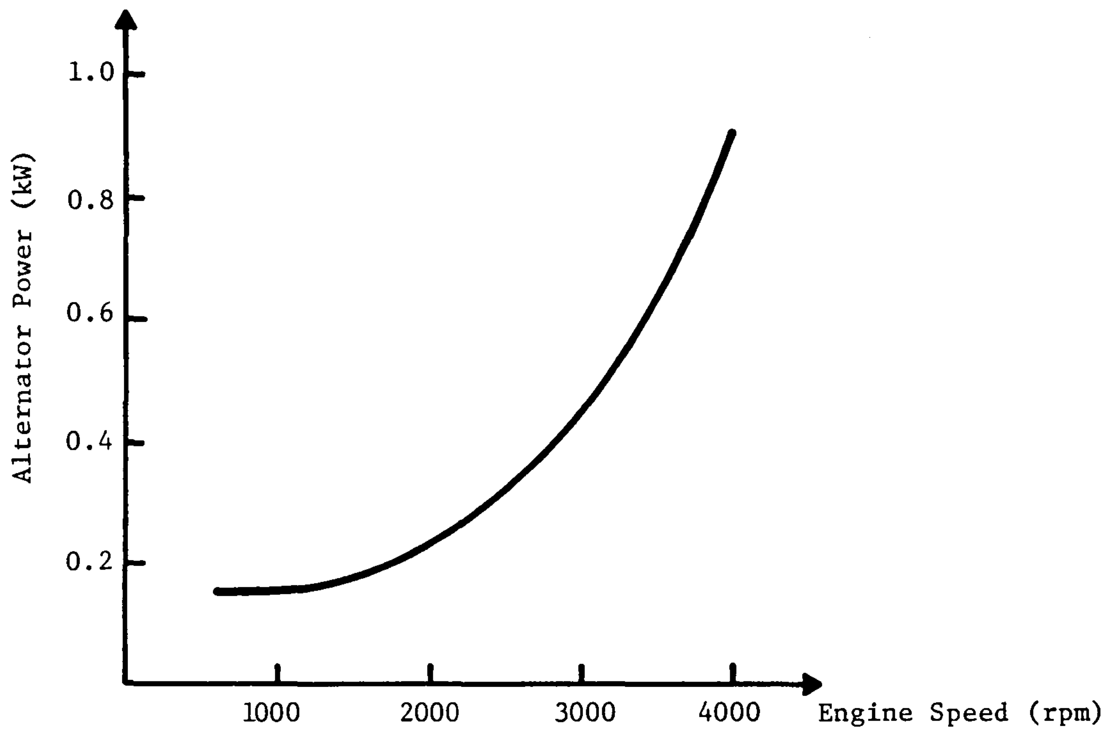

Figure 6.5-1 Alternator Power Demand

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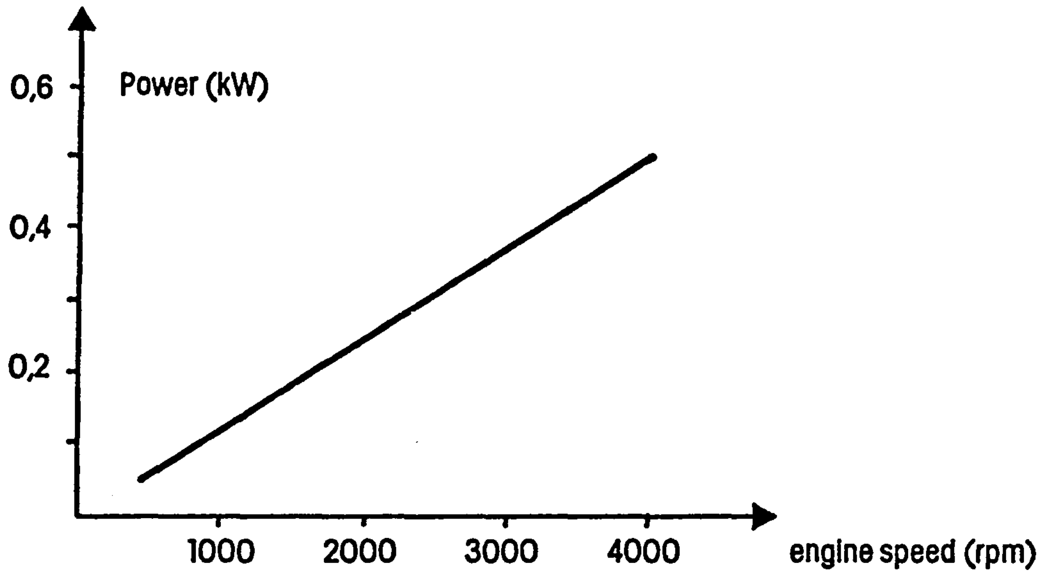

Figure 6.6-1 Power Consumption - Oil Pump

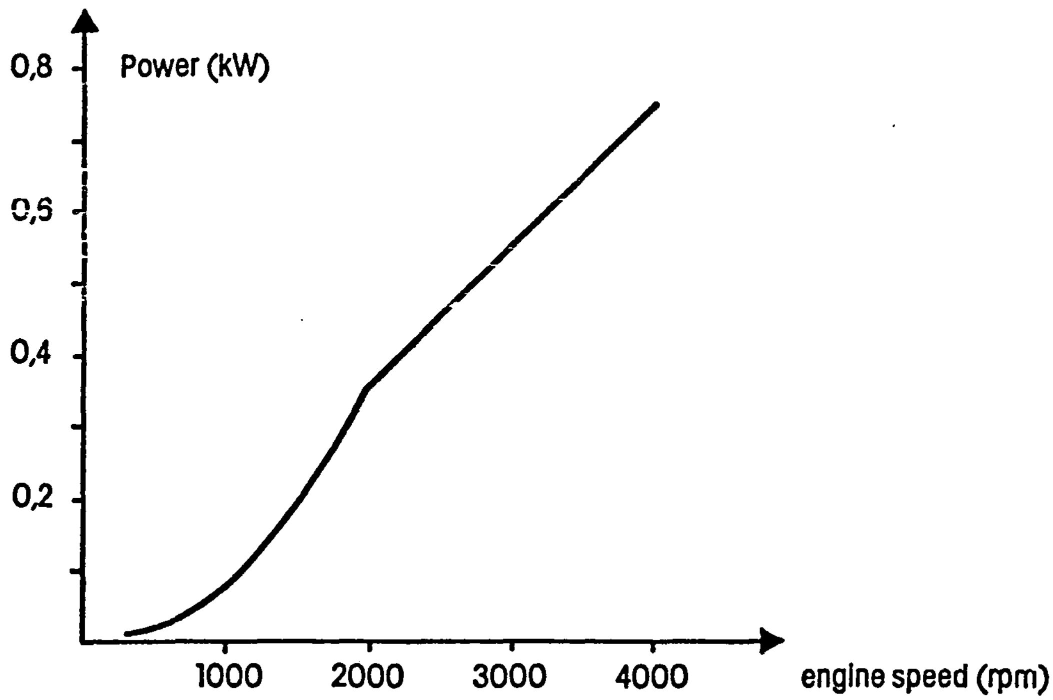

Figure 6.7-1 Power Consumption - Power Steering

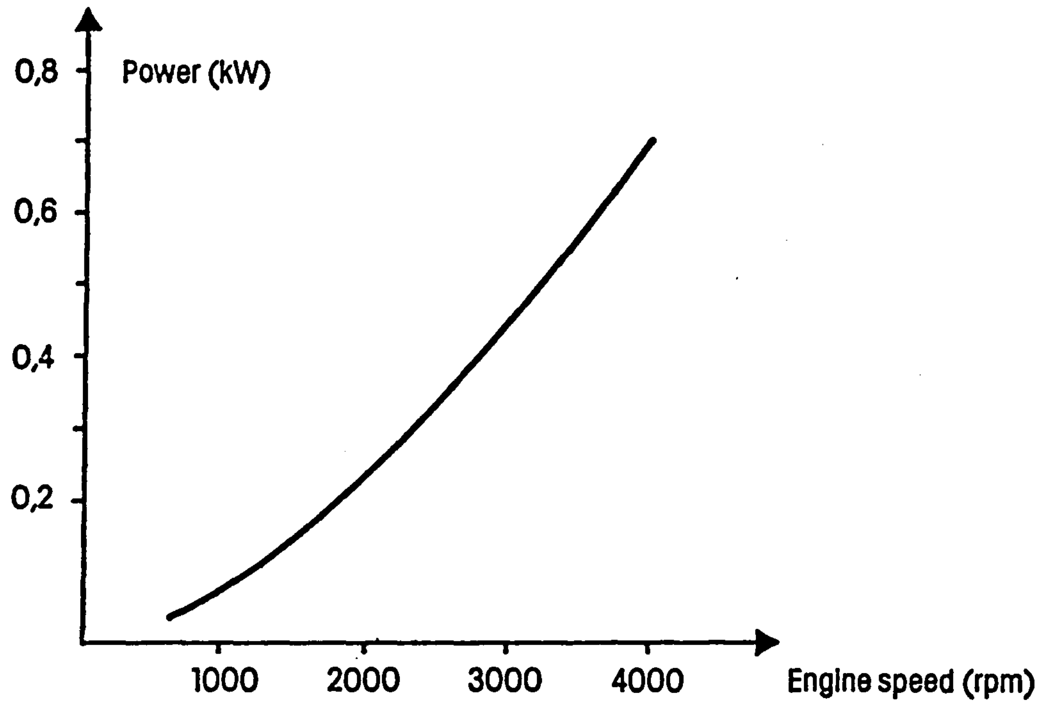

Figure 6.8-1 Hydrogen Gas Compressor Power Consumption

6.9 Radiator Fan

The radiator fan was assumed to be disengaged during the mileage calculation because ram air was sufficient for cooling.

6.10 Belt Losses

An extra power demand of 0.3 kW at 4000 rpm was assumed for the V-belts.

6.11 Total Auxiliary Power Demand

At full engine load and speed, the total auxiliary power demands are:

Combustor air blower	1.85 kW
Water pump	0.64 kW
Alternator	0.33 kW
Lubrication oil pump	0.50 kW
Hydrogen compressor	0.70 kW
Belt losses	0.30 kW
TOTAL	4.32 kW

SECTION 7.0 ENGINE INSTALLATION

7.0 ENGINE INSTALLATION

A study was performed on the packagability of the Reference Engine in a GM X-body car (Pontiac Phoenix). The compact Engine Drive System used an all gear synchronization coupled to the GM Automatic Transaxle.

The following are the two major body modifications which were necessary: 1) the radiator was moved 25 mm forward; 2) the central part of the front cross member was moved 90 mm forward.

7.1 Arrangement and Auxiliaries

The introduction of a speed increaser between the crankshafts and the drive shaft necessitated the addition of an external balancer shaft to retain the completely balanced engine concept. Further design work indicated that auxiliary units such as the water pump, the alternator, the hydraulic pump, and the fan would be easily driven using a duplex chain connection from the rear of the balancer shaft.

The incorporation of a speed increaser required a split output shaft with the front end running at engine speed synchronized with one of the crankshafts, which would drive the air blower via a variator. A higher speed was unacceptable from a reliability stand point. The balancer shaft was also connected to this end of the shaft through a 1:1 gear assembly fitted with a duplex chain.

To rotate the rear end of the drive shaft at the selected speed differential of 1.4:1, a gear drive was used, which allowed the starter motor to be connected through a gear train.

With the addition of the chain and gear drive to the front end of the drive unit, the positions of the hydrogen and oil pumps were transposed. Because of the additional crankcase profile around the balancer shaft, the burner blower housing and motor were moved forward towards the radiator, which allowed the starter motor to be located beneath it.

The water pump is driven by a duplex chain from the balancer shaft, and is coupled with a belt drive to the alternator and the power steering pump.

The placement of the auxiliaries and the vehicle installation are shown in Figures 7.1-1, 7.1-2 and 7.1-3. Incorporating the above decisions will enable the ground clearance to be maintained and allow sufficient space between the top of the engine and the hood.

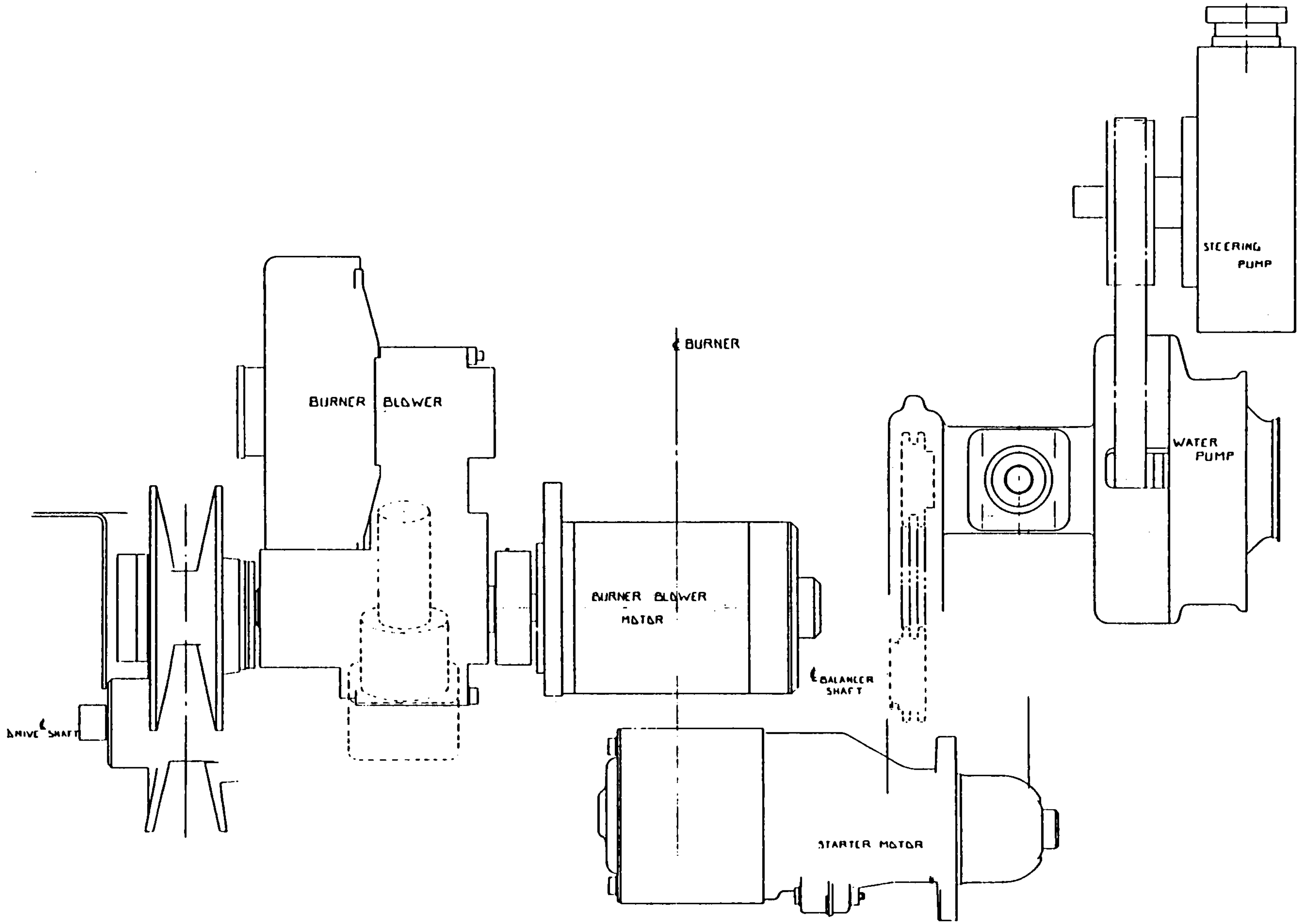

Figure 7.1-1 Water Pump Drive

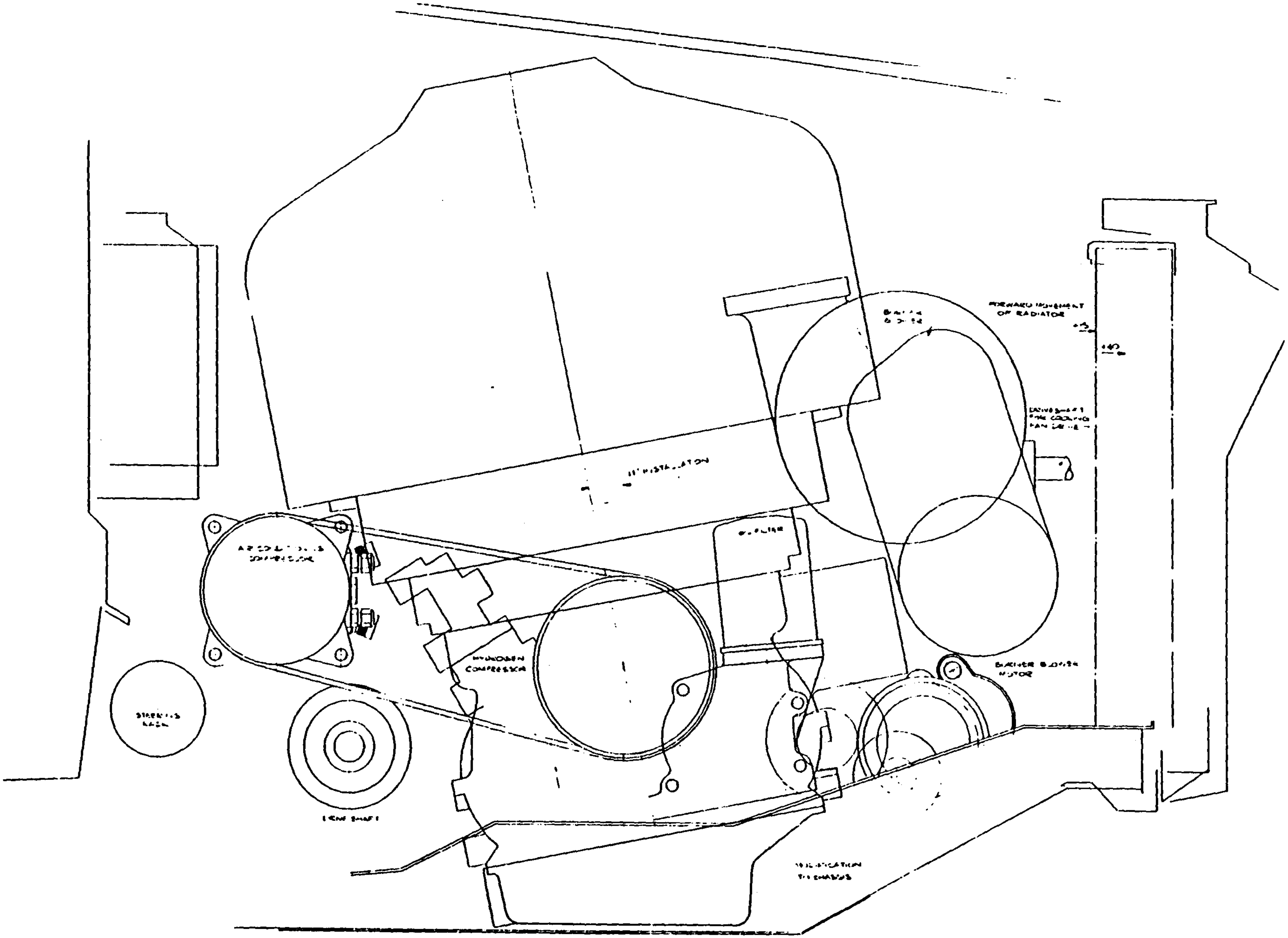

Figure 7.1-2 Engine Installation (Side View)

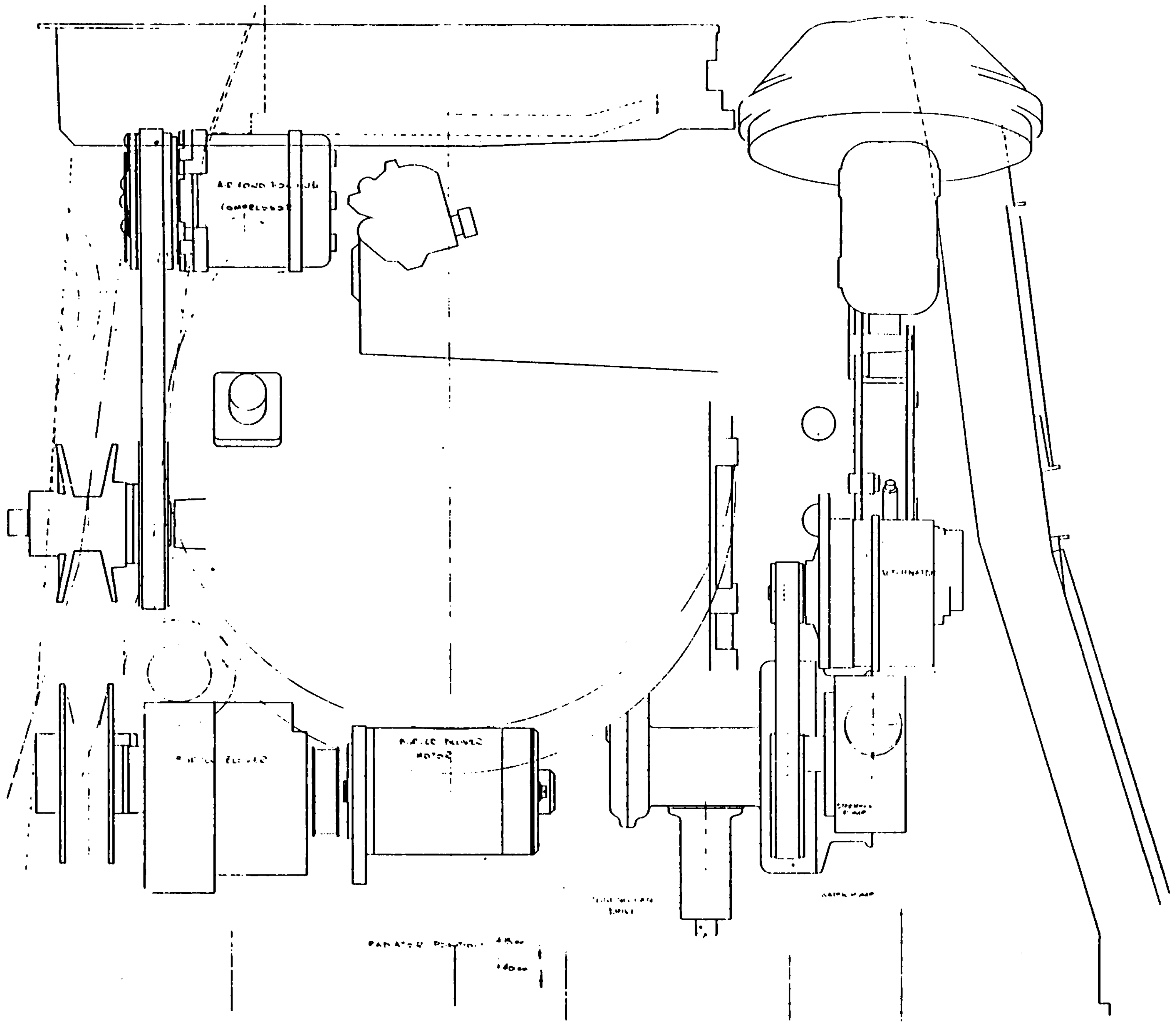

Figure 7.1-3 Engine Installation (Top View)

SECTION 8.0 OPTIMIZATION AND VEHICLE SIMULATION

8.0 OPTIMIZATION AND VEHICLE SIMULATION

8.1 Vehicle Installation Data

Vehicle installation data, such as vehicle inertia weight, road load, and transmission strongly influenced the system performance and engine design parameters.

Basic Objectives Which Influence the Engine Design

Performance Goals:	0 to 60 mph in 15 seconds 50 to 70 mph in 10.5 seconds 0 to 100 feet in 4.5 seconds
Fuel Economy Goals:	30% better than a spark ignition engine-powered automobile

Vehicle Characteristics

The current trend of automotive manufacturers is to develop highly efficient, downsized, front-wheel vehicles. The Reference Engine was designed with this in mind, as is evidenced by the selection of a 1984 Pontiac Phoenix as the Reference Vehicle. Currently, the 1981 Pontiac Phoenix is a technology leader in the domestic intermediate market. However, by 1984, it is felt that this vehicle will still be in production in its present form with minor modifications, and will exemplify a typical domestic intermediate vehicle. Therefore, the design data of the Phoenix was used for the Reference Vehicle Specification. Projected improvements to the vehicle that are in process were also included. The vehicle characteristics are summarized in Table 8.1-1.

A variable displacement gearbox oil pump, used in GM's new X-car transmission (THM 125), was added to the system. The pump power consumption with engine speed is shown in Figure 8.1-1.

	EPA CYCLES	PERFORMANCE
Vehicle Mass	3125 lbs.	3170 lbs.
Engine Inertia	6.07 lbm. ft.^{2}	6.07 lbm. ft.^{2}
Torque Converter Turbine Inertia	.54 lbm. ft.^{2}	.54 lbm. ft.^{2}
Transmission Inertia	.19 lbm. ft.^{2}	.19 lbm. ft.^{2}
Wheel Inertia	45.40 lbm. ft.^{2}	90.80 lbm. ft.^{2}
Aerodynamic Drag Coefficient	.420	.417
Rolling Friction Coefficient	.0110	.0095
Air Density	.0727 lbm/ft.^{3}	.0727 lbm/ft.^{3}
Fuel Density	6.17 lbs/gal.	6.17 lbs/gal.
Frontal Area	21.34 ft.^{2}	21.34 ft.^{2}
Idle Speed	$\sim$ 600 rpm	$\sim$ 600 rpm
Engine	RESD 308-01	RESD 308-01
Transmission Gear Ratios		
1st	2.91	2.91
2nd	1.55	1.55
3rd	1.00	1.00
4th	.71	.71
Accessory Loads	P/S, F/P	P/S, F/P
Lockup Speed	30 mph	65 mph
Drop Out Speed	25 mph	60 mph
Trasmission Efficiency	Ricardo Polynomial	Ricardo Polynomial
Response Delay	0 sec.	.2 sec.
Torque Converter	GM THM 125	GM THM 125

Table 8.1-1 Vehicle Simulation Parameters

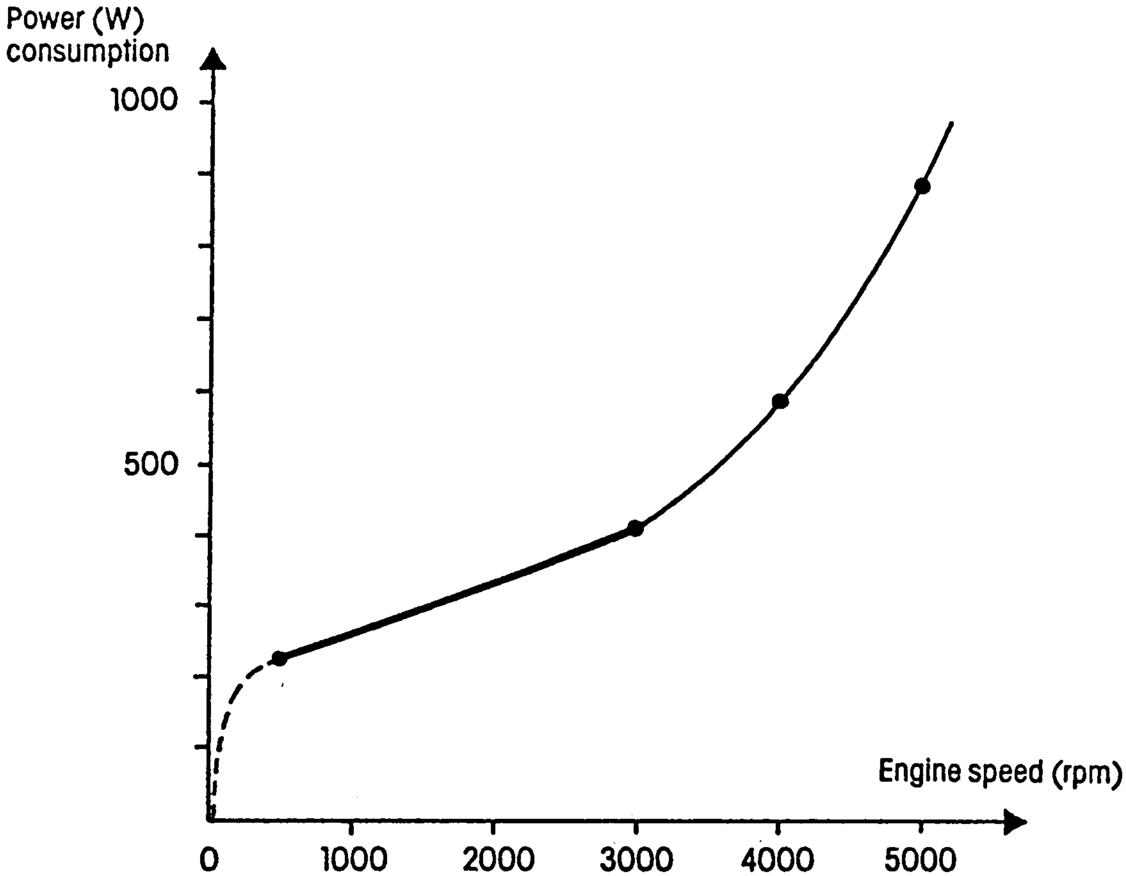

Figure 8.1-1 Pump Power Consumption with Engine Speed

Accessories

These accessories are on the vehicle: power steering, power brakes, and air conditioning.

8.2 Engine Sizing Study

Using a Vehicle Simulation Code and a technique developed by General Motors, the Reference Engine power level was studied to determine the optimum size.

The sizing technique consisted of simulating urban, highway (and thus combined mileage), and 0-60 mph time. Performance was plotted against fuel economy. For a given system, the axle ratio was varied; the acceleration and fuel economy was calculated, and those points were plotted. The engine size was then varied through scaling techniques, and similar curves with various axle ratios were calculated and plotted. Finally, a tangent line was drawn, which depicted the optimimum fuel economy/performance. This study showed that the optimum engine size for the Reference Engine is 60 kW.

During the Reference Engine sizing study, it was discovered that the GM torque converter was a serious liability for the Stirling Engine.

Figure 8.2-1 shows a comparison of three different torque converters. The size factor was plotted against the speed ratio. As is evidenced, there is considerable difference between the GM torque converter and the Chrysler unit (10-3/4"). The RESD Design II is a "paper" torque converter created to match the 60 kW Reference Engine.

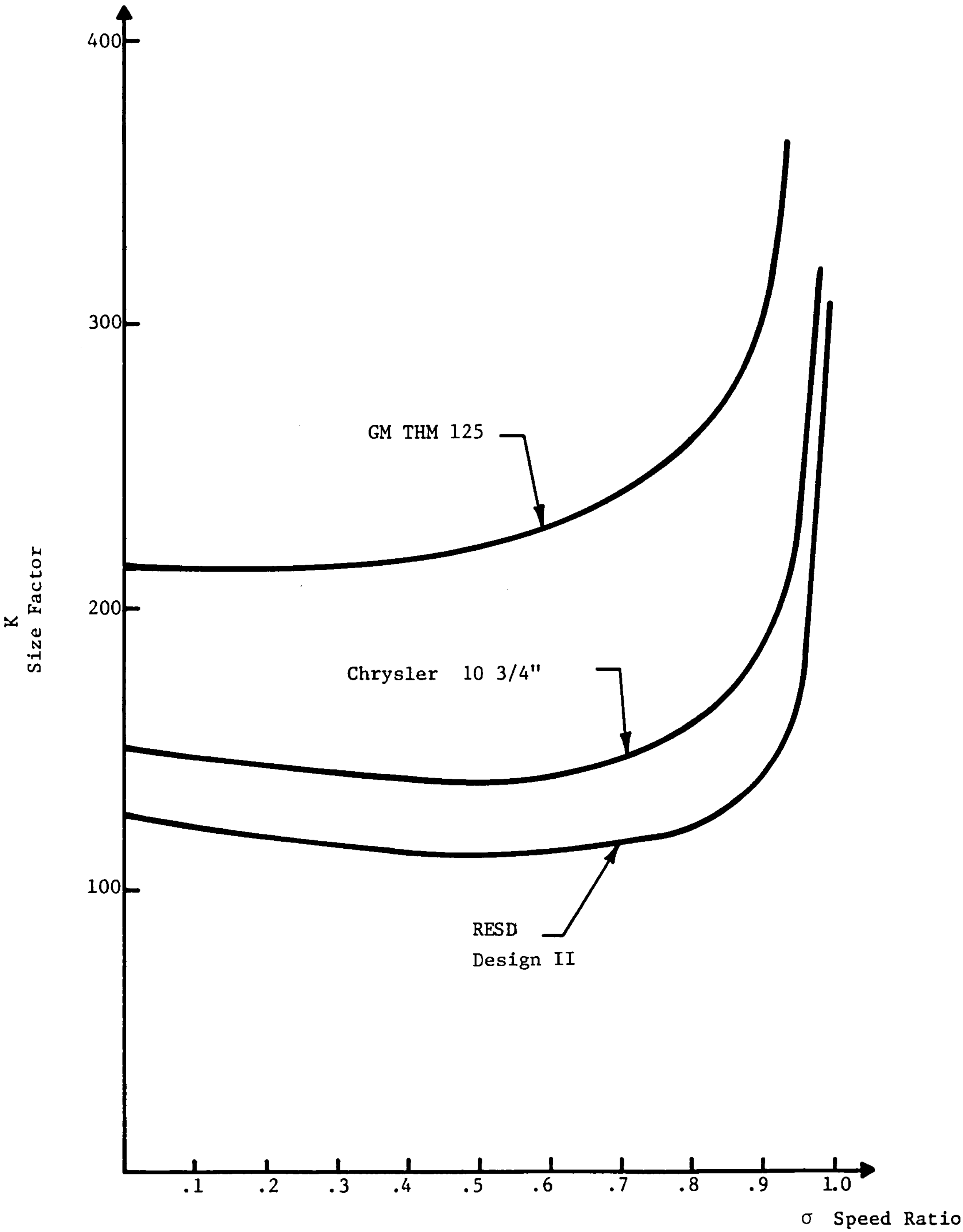

Figure 8.2-1 Torque Converter Size Factor Characteristics

Figure 8.2-2 graphically demonstrates the matching improvement of the RESD Design II. This was done by two means. First, the size of the torque converter was increased, which moved all speed ratio lines to the left (lower speed) and into the high efficiency region of the engine map. However, this also penalized idle fuel consumption. Idle output shaft torque was determined by moving up the stall line ($\sigma = 0$) to 600 rpm. To cure this, the blading of the torque converter was modified to attain a "drooping K" curve (the size factor decreased as the speed ratio increased from 0 to .5 or .6, then increased as the speed ratio increased further).

Although some automotive companies use "drooping K" converters, some feel that it adversely affects performance. As the speed ratio increases, engine speed will decrease for a short time. If the torque curve is rising with rpm at that point, as most spark ignition engines would, torque would drop which would produce a feeling of poor performance. The Reference Engine did not suffer from this effect since engine torque actually increased as speed decreased.

To achieve the proper torque converter characteristic within the confines of the transmission housing, a step-up gear was introduced between the engine and torque converter. The effect of this was to increase the "effective size" of the torque converter, as shown in Figure 8.2-3. The step-down gear after the torque converter was not needed as the axle ratio required (for optimization) was numerically too small to be accomplished in a confined planetary system. The combination of the required axle ratio and step-down gear resulted in a ratio that was consistent with existing hardware.

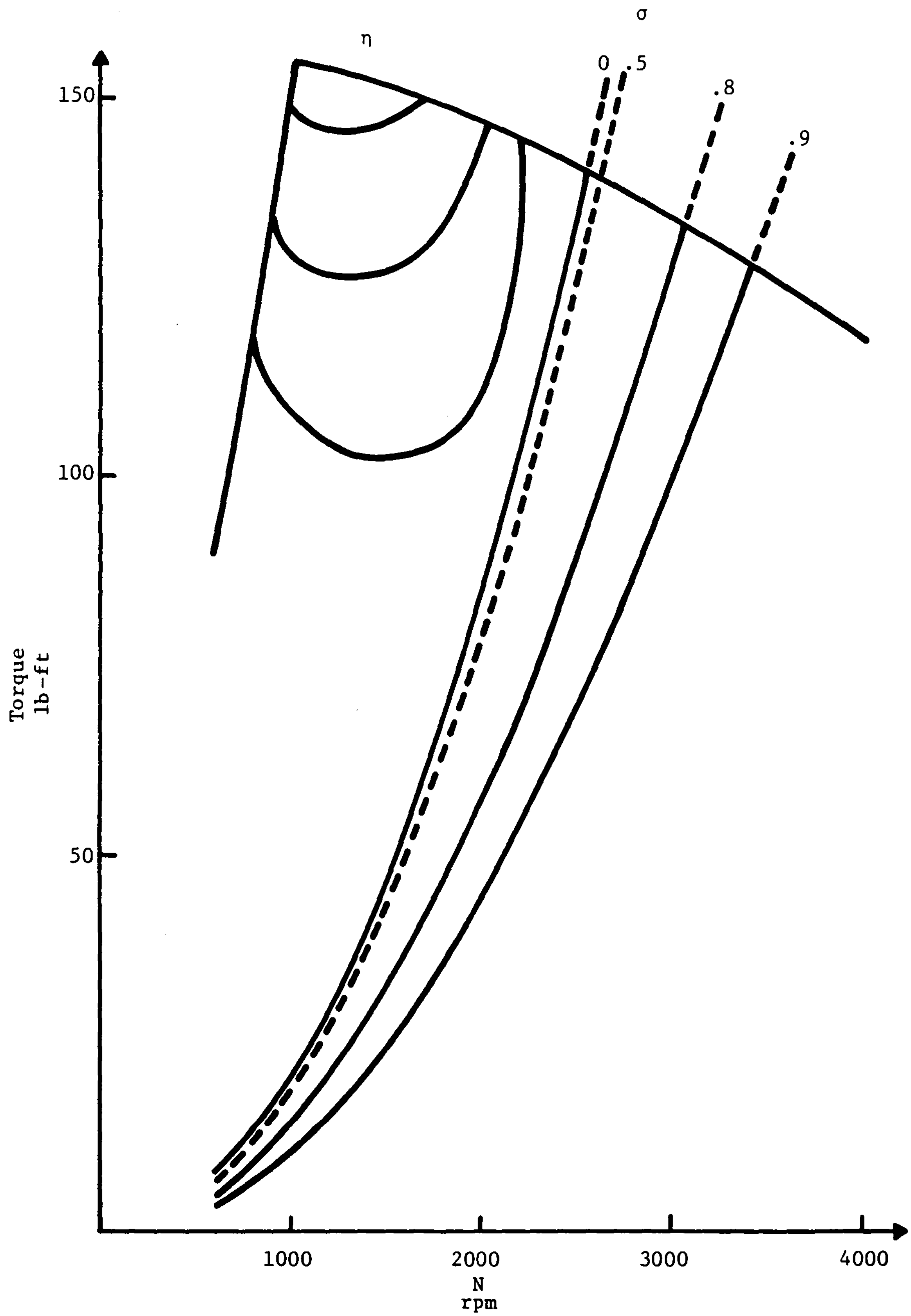

Figure 8.2-2 Engine-Torque Converter Operating Conditions
RESD - GM 9.5" T/C

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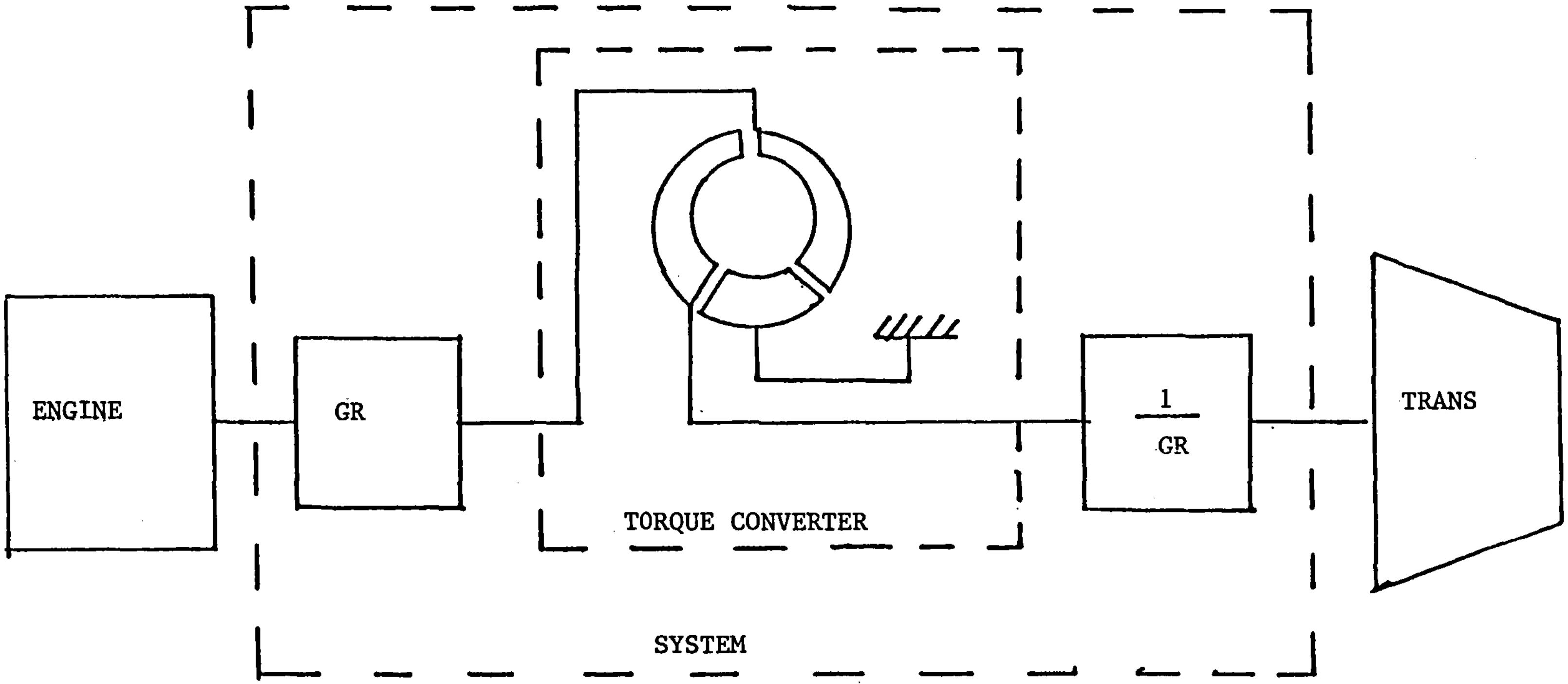


$$K_{SYS} = K_{TC} \,/\, GR^{1.5}$$

Figure 8.2-3 Effect of Step-Up Gear

8.3 Engine Reoptimization

The full-load power requirement, defined at 4000 rpm engine speed and 15 MPa cycle mean pressure, is 60 kW.

8.3.1 Definition of the Part Load Optimization Point

A 12.2 kW engine shaft power at 2000 rpm definition was used. This load point was above an average power and speed operating point of the combined CVS cycle. The reason for using an optimization part load speed higher than the calculated driving cycle simulation average engine speed was because too low a speed at the optimization point made the computer code almost ignore the pumping losses influence on engine part load efficiency (because these losses are approximately proportional to speed^3). Hence, the resulting engine would have very large pumping losses at full-speed.

This means that a relative error in the pressure drop coefficient would give an error in the net shaft output which increases with decreasing part load speed. In other words, the confidence in the full-load power level depends on the chosen part-load speed. A compromise, which gave a reasonable ratio between pumping losses and indicated full-load power, was a speed around 2000 rpm. This speed still gives the majority of the benefit of part-load optimization compared to full-load optimization.

8.3.2 Recapitulation of Basic Operational Data

Heater Tube Outside Wall Temperature	820°C
Coolant Top Tank Temperature	50°C
Maximal Cycle Mean Pressure	15 MPa
Full Load Engine Speed	4000 rpm
Working Gas	Hydrogen

8.3.3 The Reoptimized Engine

The optimization code was executed with the above conditions.

The calculated full-load power is 60.1 kW, which is within the computer program iteration stop-test tolerance.

Geometrical and operational data for the RESD are shown in Tables 8.3.3-1 and 8.3.3-2. Subsystem efficiencies are shown in Figures 8.3.3-1 through 8.3.3-4. The heat flows are shown in Table 8.3.3-3 and the performance maps are shown in Figures 8.3.3-5 and 8.3.3-6. Figure 8.3.3-7 shows the External Heat System efficiency versus fuel mass flow for the engine.

8.4 Reference Engine System Design (RESD) Vehicle System Analyses

This section presents the results of the vehicle simulation and optimization studies performed on various RESD systems. The basic tool used in this study was a Vehicle Simulation Program, which was a growth version of the University of Wisconsin program. MTI has tailored this program to the specific needs of Stirling-powered vehicles, while retaining its utility as a general heat engine vehicle simulation program.

Quantity	Unit	RESD Optimization
Drive Mechanism, Cylinders		
Piston diameter	mm	63.0
Piston rod diameter	mm	12.6
Displacer dome height	mm	120.0
Gap displacer dome-cylinder wall	mm	*
Crank radius	mm	17.0
Stroke	mm	34.0
Connecting rod length	mm	95.0
Swept volume	cm^3	106.0
Regenerator (Gauze type)		
Units per cycle	-	1
Diameter	mm	64.0
Top cross section area	cm^2	32.2
Length	mm	54.3
Wire diameter	m	*
Filling factor	%	*
Weight per engine	kg	1.8
Cooler		
Units per cycle	-	1
Tubes per cycle	-	*
Inside tube diameter	mm	*
Outside tube diameter	mm	*
Length of one tube	mm	*
Effective length of one tube	mm	*
Heater		
Tubes per cycle	-	20
Inside tube diameter	mm	*
Outside tube diameter	mm	*
Length of one tube	mm	*
Effective length of one tube	mm	*
Connecting Ducts		
Comp. cylinder - cooler	cm^3	*
Cooler - regenerator	cm^3	*
Regenerator - heater	cm^3	*
Heater - exp. cylinder	cm^3	*

*"Protectible Data"...not furnished in this report.

Table 8.3.3-1 Geometry of the Reference Engine

Note: In the friction was included 200 W oil seal loss at 4000 rpm and in the auxiliaries was included 300 W V-belt loss at 4000 rpm, both proportional to the engine speed.

FULL LOAD POINT	p = 15 MPa n = 4000 rpm
Indicated Power	73.3 kW
Friction	9.6 kW
Auxiliaries	3.6 kW
Net Power	60.1 kW
External Heating Efficiency	90.5 %
Net Efficiency	34.2 %

PART LOAD POINT	p = 5 MPa n = 2000 rpm
Indicated power	15.0 kW
Friction	2.0 kW
Auxiliaries	0.8 kW
Net Power	12.2 kW
External Heating Efficiency	91.7 %
Net Efficiency	37.7 %

MAXIMUM EFFICIENCY POINT	p = 15 MPa n = 1100 rpm
Indicated Power	24.8 kW
Friction	2.2 kW
Auxiliaries	0.5 kW
Net Power	22.1 kW
External Heating Efficiency	92.4 %
Net Efficiency	43.5 %

LOW LOAD POINT	p = 5 MPa n = 1000 rpm
Indicated Power	7.9 kW
Friction	0.9 kW
Auxiliaries	0.4 kW
Net Power	6.6 kW
External Heating Efficiency	89.8 %
Net Efficiency	36.4 %

Table 8.3.3-2 Performance of the Reference Engine

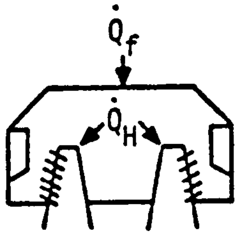
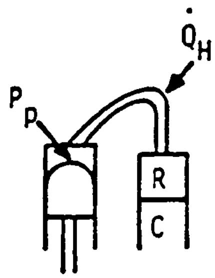
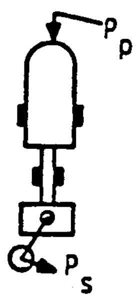
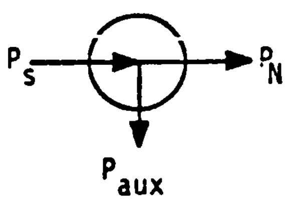
SUBSYSTEM	BOUNDARIES	EFFICIENCY %	DEFINITION
EXTERNAL HEAT SYSTEM		90.5	$\frac{\dot{Q}_{HEATER}}{\dot{Q}_{FUEL}}$
ENGINE		46.1	$\frac{P_{PISTON}}{\dot{Q}_{HEATER}}$
DRIVE and SEALS		87.0	$\frac{P_{SHAFT}}{P_{PISTON}}$
AUXILIARIES and CONTROLS		94.2	$\frac{P_{NET}}{P_{SHAFT}}$
		34.2	$\frac{P_{NET}}{\dot{Q}_{FUEL}}$

Figure 8.3.3-1 Subsystem Efficiencies at Full Load (p = 15 MPa, 4000 rpm)

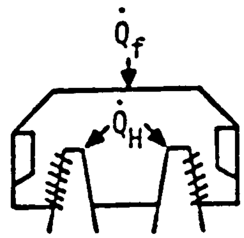
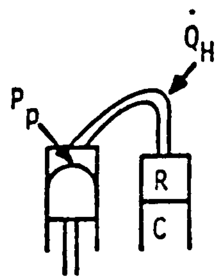
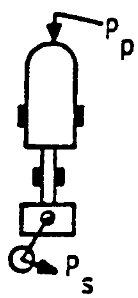
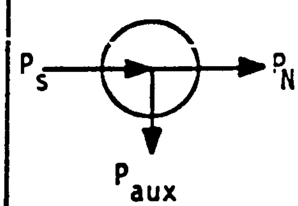
SUBSYSTEM	BOUNDARIES	EFFICIENCY %	DEFINITION
EXTERNAL HEAT SYSTEM		91.7	$\dfrac{\dot{Q}_{HEATER}}{\dot{Q}_{FUEL}}$
ENGINE		50.9	$\dfrac{P_{PISTON}}{\dot{Q}_{HEATER}}$
DRIVE and SEALS		86.8	$\dfrac{P_{SHAFT}}{P_{PISTON}}$
AUXILIARIES and CONTROLS		93.1	$\dfrac{\bar{P}_{NET}}{P_{SHAFT}}$
		37.7	$\dfrac{P_{NET}}{\dot{Q}_{FUEL}}$

Figure 8.3.3-2 Subsystem Efficiencies at Part Load (p = 5 MPa, 2000 rpm)

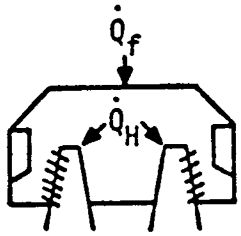
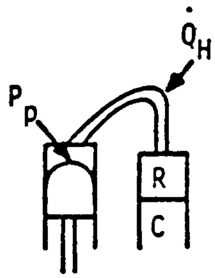
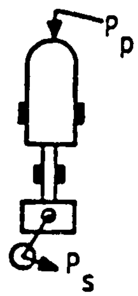
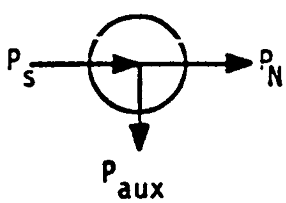
SUBSYSTEM	BOUNDARIES	EFFICIENCY %	DEFINITION
EXTERNAL HEAT SYSTEM		92.4	$\frac{\dot{Q}_{HEATER}}{\dot{Q}_{FUEL}}$
ENGINE		52.8	$\frac{P_{PISTON}}{\dot{Q}_{HEATER}}$
DRIVE and SEALS		91.1	$\frac{P_{SHAFT}}{P_{PISTON}}$
AUXILIARIES and CONTROLS		97.9	$\frac{P_{NET}}{P_{SHAFT}}$
		43.5	$\frac{P_{NET}}{\dot{Q}_{FUEL}}$

Figure 8.3.3-3 Subsystem Efficiencies at Maximum Efficiency (p = 15 MPa, 1100 rpm)

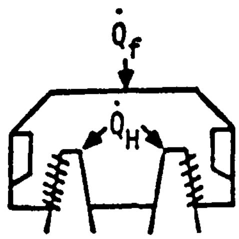
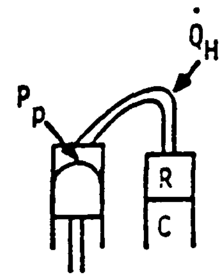
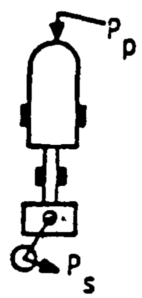
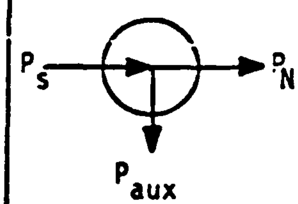
SUBSYSTEM	BOUNDARIES	EFFICIENCY %	DEFINITION
EXTERNAL HEAT SYSTEM		89.8	$\frac{\dot{Q}_{HEATER}}{\dot{Q}_{FUEL}}$
ENGINE		48.5	$\frac{P_{PISTON}}{\dot{Q}_{HEATER}}$
DRIVE and SEALS		89.1	$\frac{P_{SHAFT}}{P_{PISTON}}$
AUXILIARIES and CONTROLS		93.8	$\frac{P_{NET}}{P_{SHAFT}}$
		36.4	$\frac{P_{NET}}{\dot{Q}_{FUEL}}$

Figure 8.3.3-4 Subsystem Efficiencies at Low Load (p = 5 MPa, 1000 rpm)

The Heat Flows in Reference Engine

$\dot{Q}_f$ = Heat from the fuel

$\dot{Q}_{ex}$ = Heat losses in heat generating system

$\dot{Q}_E$ = Heat to the heater head

$\dot{Q}_{cb}$ = Heat losses in cylinders and regenerators

$\dot{Q}_C$ = Rejected heat in the cycle coolers

P_i = Indicated power

A = Full load, p = 15 MPa, n = 4000 rpm

B = Part load, p = 5 MPa, n = 2000 rpm

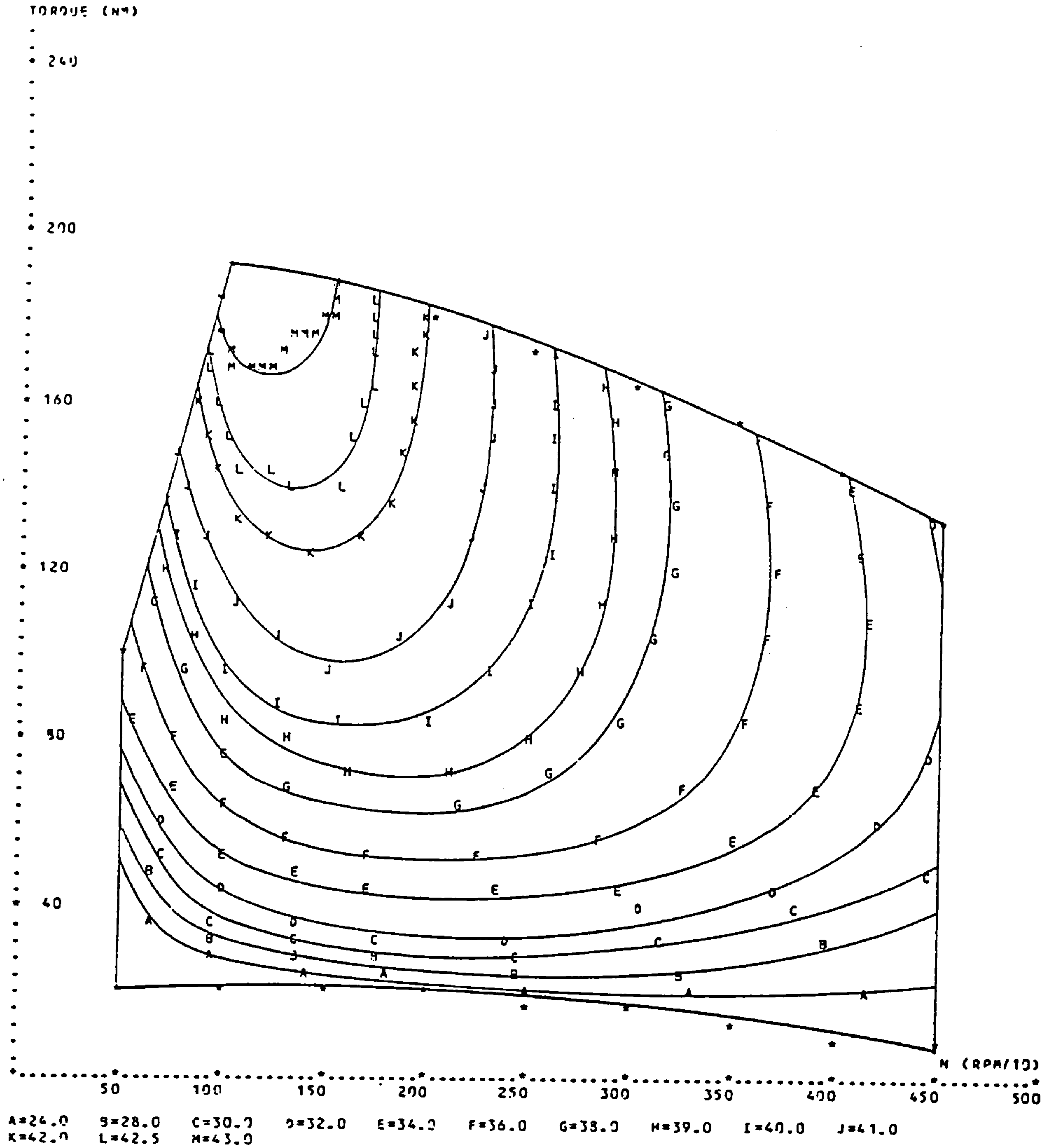
C = Maximum efficiency load, p = 15 MPa, n = 1100 rpm

D = Load low, p = 5 MPa, n = 1000 rpm

All values in kW

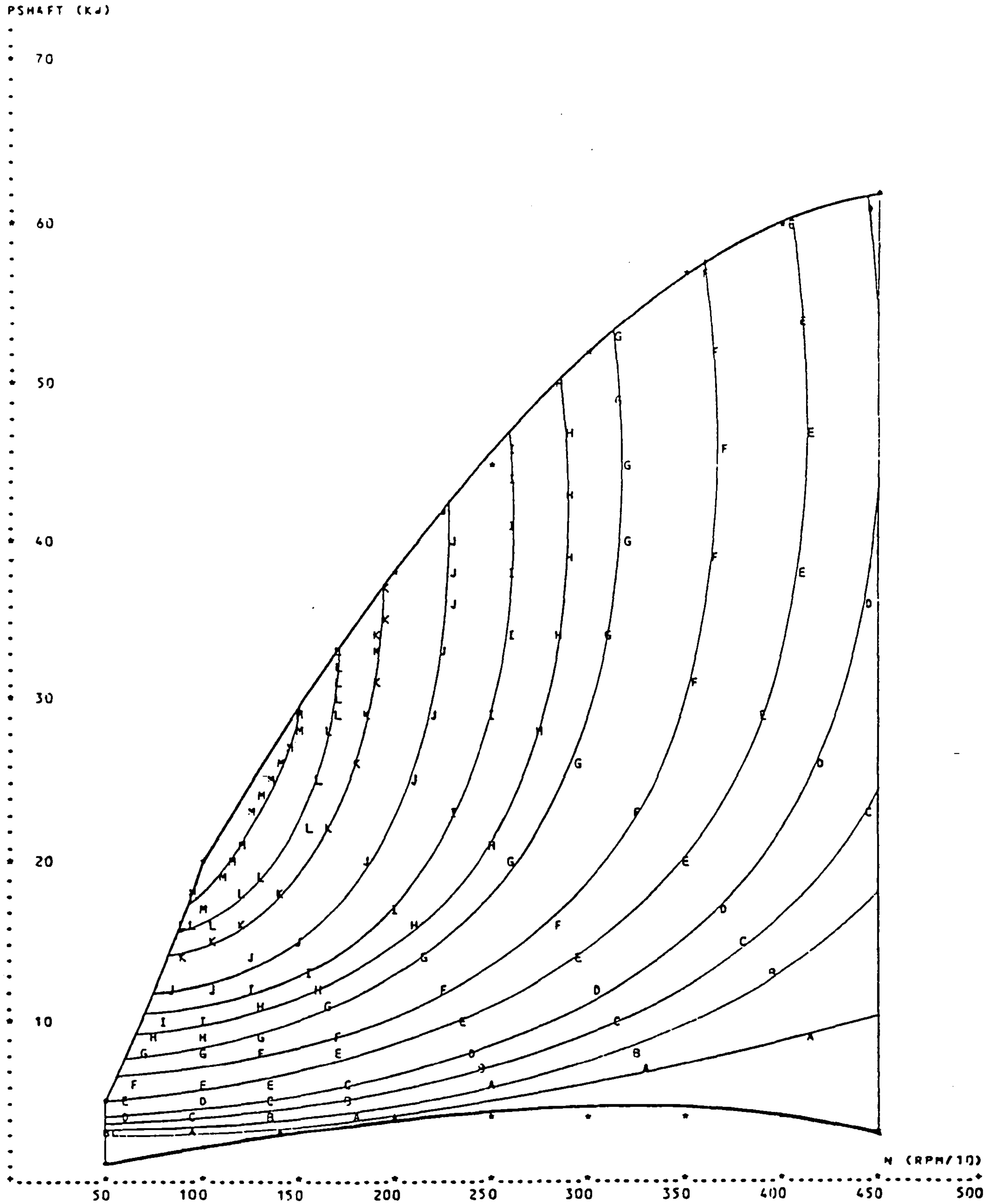
	A	B	C	D
$\dot{Q}_f$	175.8	32.2	50.7	18.2
$\dot{Q}_{ex}$	16.0	2.6	3.8	1.8
$\dot{Q}_e$	159.1	29.6	46.9	16.4
$\dot{Q}_{cb}$	2.2	2.4	2.3	2.4
$\dot{Q}_c$	85.8	14.5	22.1	8.4
P_i	73.3	15.0	24.8	7.9

Table 8.3.3-3 Heat Balance



PERFORMANCE MAP OF THE REFERENCE ENGINE : CURVES OF CONSTANT NET EFFICIENCY (%)

Figure 8.3.3-5 Performance Map with Net Shaft Torque versus Speed



A=24.0 B=28.0 C=30.0 D=32.0 E=34.0 F=36.0 G=38.0 H=39.0 I=40.0 J=41.0
K=42.0 L=42.5 M=43.0

PERFORMANCE MAP OF REFERENCE ENGINE: CURVES OF CONSTANT NET EFFICIENCY (%)

Figure 8.3.3-6 Performance Map with Net Shaft Power versus Speed

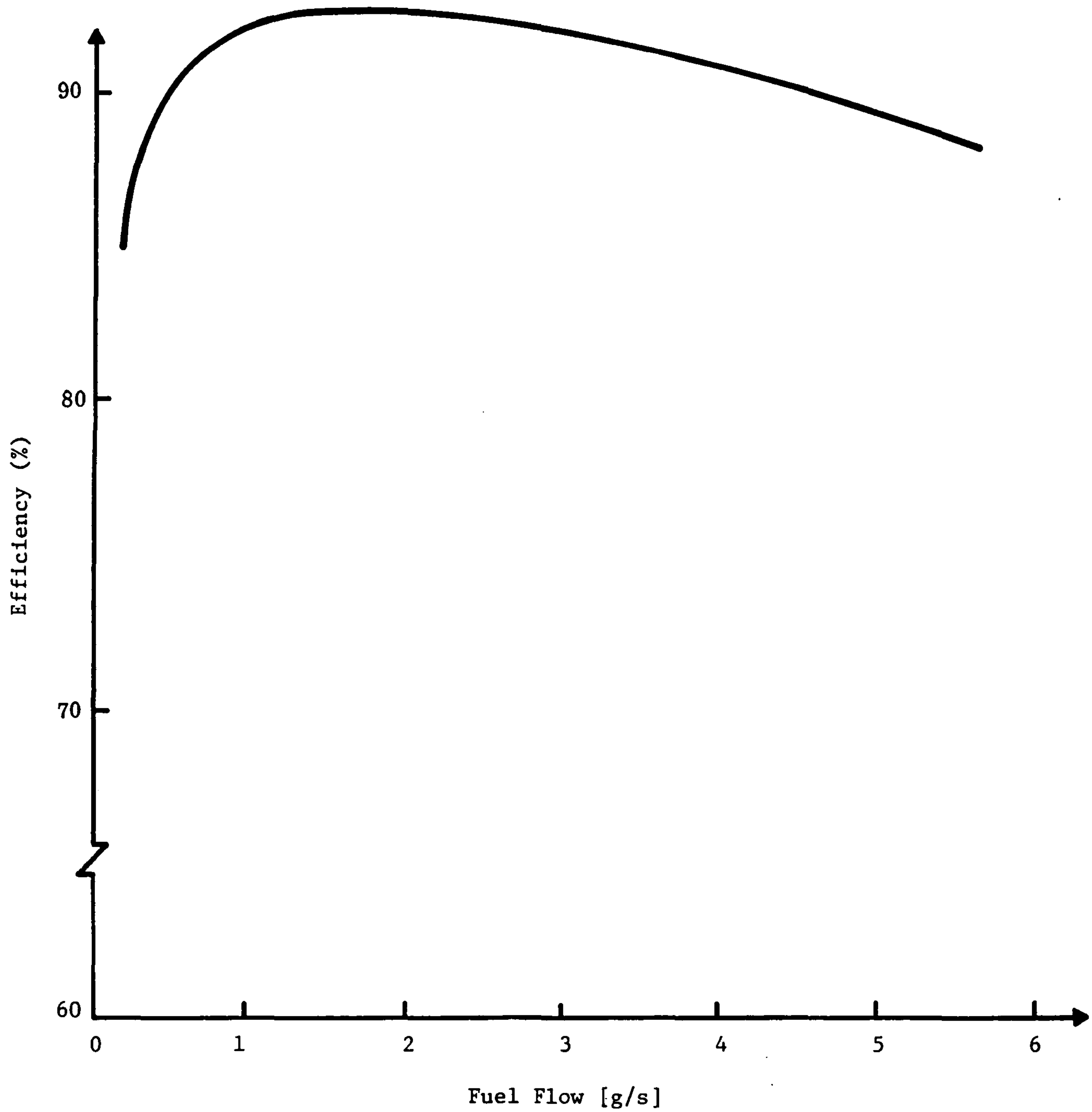


Figure 8.3.3-7 External Heat System Efficiency Versus Fuel Flow

The program was validated for spark ignition vehicles by the University of Wisconsin. Further validation comparisons were performed on a 1981 Pontiac Phoenix vehicle and on a Spirit vehicle powered by a P-40 Stirling engine. Results of the 1981 Pontiac Phoenix validation are shown in Table 8.4-1.

8.4.1 Definitions for the Mileage Calculations

8.4.1.1 The Driving Cycles

The mileage refers to the EPA combined cycle, which consists of one urban and one highway cycle, both defined in the Code of Federal Regulations (CFR).

With M and H representing the mileage of the urban cycle and highway cycle, respectively, the mileage (M-H) of the combined cycle was defined by

$$M\text{-}H = \left(\frac{0.55}{M} + \frac{0.45}{H}\right)^{-1}$$

8.4.1.2 Fuel

Heating value: 43.03 MJ/kg (18500 Btu/lb)

Density: 736 kg/m^3

Vehicle: 1981 Pontiac Phoenix

Configuration: 2.5l I4, 3 Speed Automatic, P/S, P/B, A/C

	Results	
	Calculated	Actual
Estimated mpg	22.6	22*
Highway mpg	32.7	31*
Combined mpg	26.2	26*
0-60 mph Time	14.1	12.1, 15.0, 15.4**

* Mileage numbers rounded by EPA

**Acceleration times vary greatly from source to source

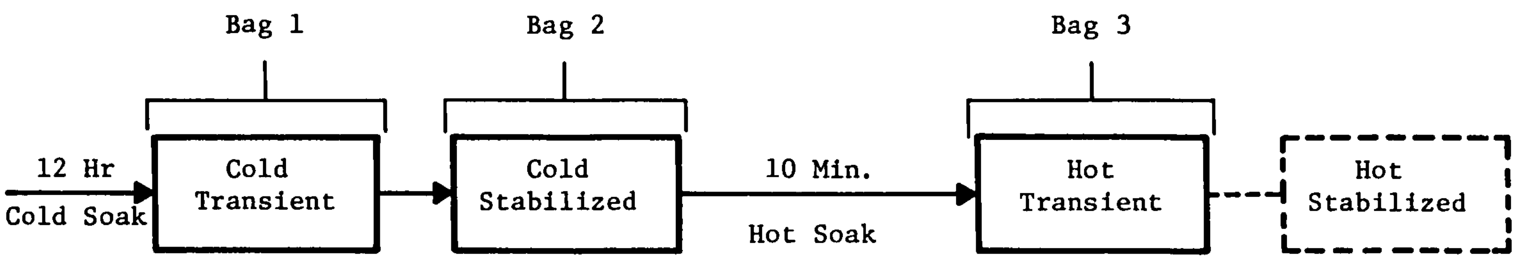
Table 8.4-1 Vehicle Simulation Code Validation

8.4.1.3 Cold Start Penalty

The cold start penalty incurred on the CVS cycle is a complex concept dealing with deviation of engine and drivetrain performance due to non-stabilized engine initial operating conditions. The concept is far too complex to handle analytically. However, due to its substantial effect on mileage predictions, it is imperative to quantify the penalty reasonably accurately.

Historically, the penalty has been assumed to be roughly equivalent to the heat stored in the hot parts of the engine. This, however, was not quite accurate due to several effects. First, the engine, transmission, and axle oil added to the penalty due to its increased viscosity at low temperatures. Second, the conduction loss mechanism was contained within the steady state maps. During the heating of the engine, much of the heat went into establishing the temperature profile that drove the conduction losses. Thus, the conduction losses were lower, which somewhat offset the energy required to heat the engine.

In order to truly quantify the cold start penalty, a test was devised to measure it directly. The urban cycle consisted of a 12 hour cold soak and 505 seconds of "cold transient" driving. This is shown schematically in Figure 8.4.1.3-1 as well as the equations which quantify the mileage. If this test is rerun without the cold soak or hot soak (starting with a stabilized idle condition), the cold start penalty can be quantified (Refer to Figure 8.4.1.3-2).

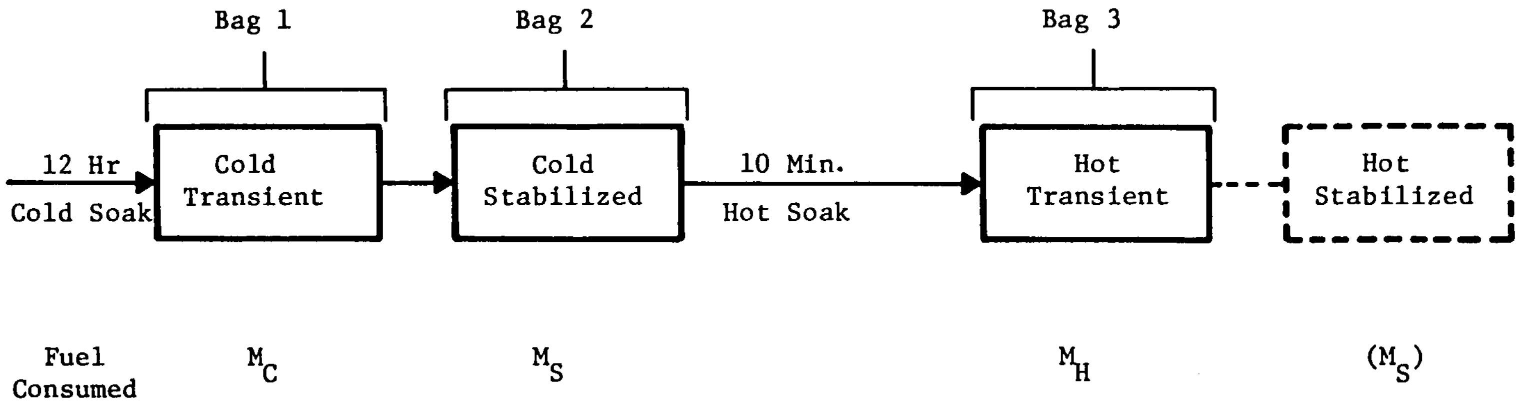


Fuel Consumed: M_C M_S M_H (M_S)

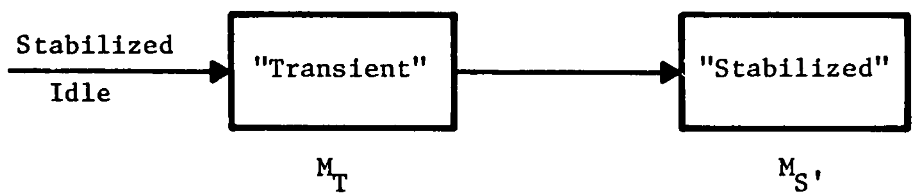
$$\text{Estimated mpg} = \frac{\text{Distance}}{.43(M_C+M_S) + .57(M_H+M_S)} = \frac{\text{Distance}}{.43M_C + .57M_N + M_S}$$

Figure 8.4.1.3-1 Urban Driving Cycle Test

811684



$$\underline{\text{Estimated} = \frac{\text{Distance}}{43(M_C+M_S) + .57(M_H+M_S)} = \frac{\text{Distance}}{.43M_C + .57M_H+M_S}}$$
$$\text{mpg Cold Start Penalty Test}$$



$$\text{"Cold Start" Penalty} = .43(M_C-M_T) + .57(M_H-M_T) + (M_S-M_{S'})$$

$$\underset{\text{mpg}}{\text{Estimated}} = \frac{\text{Distance}}{\text{CSP} + M_T + M_{S'}}$$

Figure 8.4.1.3-2 Cold Start Penalty Test

811685

The cold start penalty consists of engine heating losses, and viscous losses in the engine, transmission, and axle due to initially cold oil. To quantify the cold start penalty of the Reference Engine based on P-40 data, one must segregate the losses due to different mechanisms and adjust them appropriately (Figure 8.4.1.3-3). Tables 8.4.1.3-1 and 8.4.1.3-2 describe this process.

In general, the engine heating loss was assumed to be proportional to the ratio of stored heat in the engine. The reheating loss was assumed to be proportional to the stored heat and it was also assumed that the run-down losses were cut in half by completely closing the air throttle after the ignition was turned off.

The engine oil penalty was considerably less due to the efficiency of the Reference Engine and due to the synthetic low viscosity oil that was used. Transmission losses were about equal; however, the axle losses were much lower due to the use of an efficient planetary gear set lubricated by automatic transmission fluid rather than hypoid gears lubricated by SAE 90 weight oil.

It was recognized that several assumptions were made that have little backing; however, these assumptions did not have a dramatic effect on the mileage, as they were made in pairs, (i.e., subtraction, then addition). The methodology is important because it establishes a framework to build upon.

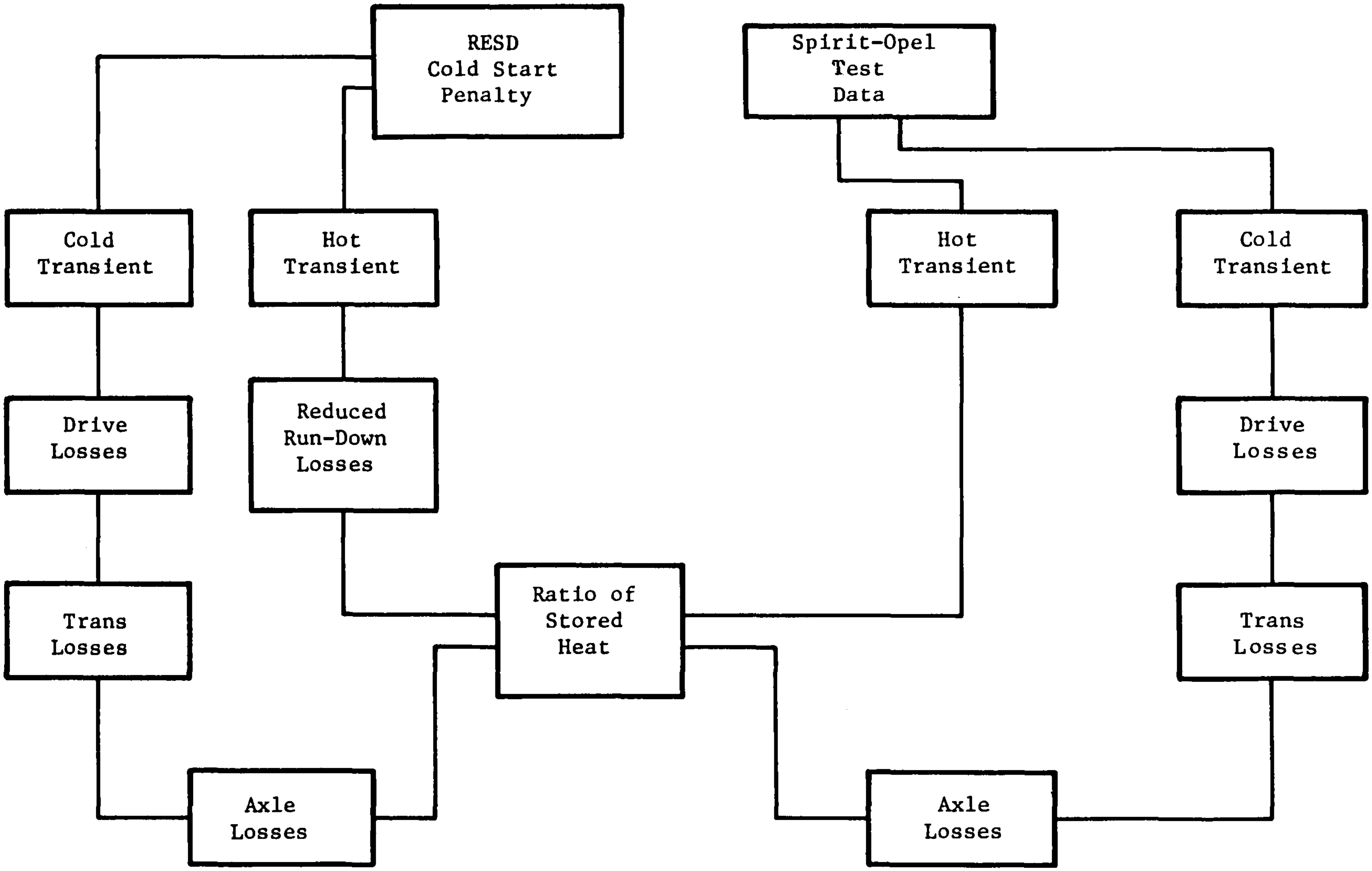


Figure 8.4.1.3-3 RESD Cold Start Penalty

811686

	Cold Transient	Hot Transient	Total
	80.8	46.9	128
	86.6	68.4	155
	107.9	74.4	182
	122.0	45.8	168
$\bar{X}$	99.3	58.9	158.2

Cold Transient

Engine Oil Penalty	
Complete Cold Transient @ 20°C	56 g
Oil Temperature will Rise Apply (1/2)	28 g
Transmission Losses	135 hp-sec
@ $BSFC_{(Eng. + Acc.)} = .8$ lb/hp-hr	13.6 g
Assume +50% Loss	6.8 g
Axle Losses (Hypoid with SAE 90)	390 hp-sec
@ $BSFC_{(Eng. + Acc.)} = .8$ lb/hp-hr	39.2 g
Assume +100% Loss	39.2 g
Cold Transient Engine Start Loss	99.3/.43 - 28 - 6.8 - 39.2 = 156.9

Table 8.4.1.3-1 Cold Start Penalty P-40 Test Data - Spirit and Opel

	Cold Transient	Hot Transient
Engine Losses (P-40)	156.9	103.3
Ratio of Stored Heat RESD/P-40 = .9)	141.2	93.0
Drive Losses (Oil @ 20°C = 7g)	3.5	
Transaxle Loss	13.6	
Reduced Run-Down (1/2)		46.5
Subtotal	158.3	46.5
Weighting Factors	68.0	26.5
Total CSP = 68.0 + 26.5 =	94.5 g	

Table 8.4.1.3-2 Cold Start Penalty for RESD

8.4.2 Vehicle Simulation Results

The Reference Vehicle System was modelled on a Vehicle Simulation Code and the mileage and acceleration were evaluated. The axle ratio was varied to achieve the 0-60 mph target of 15.0 seconds, the optimum axle being 3.77 with a 1.42 step-up gear. The equivalent ratio was 2.65. The 2.65 overall was held constant and the step-up gear was reduced to 1.38 with a resulting loss in mileage of 0.1 mpg. Vehicle simulation results are given in Figure 8.4.2-1.

8.5 Coolant Temperature Impact on Mileage

Computer simulations with dynamic top tank coolant temperatures were not performed but the driving cycle program is being modified to do this. Before these simulations are run, neither the influence of adding glycol to the coolant, or the influence of the cooling fan can be predicted. Before these simulations are performed, the coolant average temperature is based only on assumptions.

A computer simulation was run which was identical to previous simulations except that the coolant top tank temperature was set at 90°C instead of 50°C for the combined metro-highway cycle. (This was unrealistically high for the cold transient phase.) The resulting fuel economy decreased from 40.2 mpg to 38.2 mpg. This meant that the combined metro-highway fuel economy at a coolant top tank temperature of 70°C would yield about 39.2 mpg. A preliminary study indicated that a 50% glycol/water coolant would raise the top tank temperature by about 10°C.

Urban Mileage	
Without CSP	39.2
With CSP	33.3
Highway Mileage	57.4
Combined Mileage	
Gas	41.1
Diesel	47.3
55 MPH Cruise	
Gas	61.0
Diesel	70.0
0-60 MPH Acceleration	15 sec.
50-70 MPH Acceleration	9.6 sec.
0-100 ft. Acceleration	4.5 sec.
30% Grade	Yes

Figure 8.4.2-1 Vehicle Performance Results

SECTION 9.0 ENGINE MATERIALS

9.0 ENGINE MATERIALS

Table 9.0-1 contains a listing of the materials used in the main engine components. Table 9.0-2 shows the composition of alloys which are referred to in Table 9.0-1. Table 9.0-3 shows the composition of alloys presently used or proposed for use in the engine.

COMPONENT	MATERIAL	REMARKS
External Heat System		
Air Preheater Matrix	SIS 2361 = ANSI 310	0.1 mm Sheet
Air Preheater Housing	SIS 2361, Outer Sheet Al	
Insulation	Alumina-silicate Fiber Material	Vacuum Formed
Combustor	Nimonic 75	Coating May Be Needed
Ejectors	SIS 2361	
CGR - Valve	SIS 2361	Probably Coated With Stellite
Fuel Nozzles	SIS 2361	
Flame Shield	Alumina-Silicate Fiber Material	Vacuum Formed
Hot Engine System		
Cylinders and Regenerator Housings	Climax Molybdenum XF818	Castings
Heater Tubes	SANDVIK 12 RN 72	Seamless
Tube Fins	SIS 2361	
Regenerators	SIS 2352 ≈ AISI 304L	Gauze
Coolers	Aluminum	

Table 9.0-1 Materials Used in Main Engine Components

COMPONENT	MATERIAL	REMARKS
Cold Engine System		
Water Jacket	Aluminum	
Cold Connecting Duct Plate	SIS 0737, Nodular Cast Iron	
Piston Dome	Inconel 718	
Dome Base	SIS 2225-05	Hardened and Tempered
Piston	SIS 2541-02	Hardened and Tempered
Piston Rings	Rulon LD	
Piston Rod	SIS 2940-03	Nitrided
Piston Rod Seal		
- Housing	ASTM 6351	
- Seal Elements	Rulon LD	
Engine Drive System		
Crank Case	Aluminum	
Bed Plate	Aluminum	
Crankshafts	BS SG 37/2	Nodular Cast Iron
Connecting Rods	BS SG Iron	Nodular Cast Iron
Crosshead	BS LM 13-WP	Al 32-T65 Al Alloy

BS = British Standard

SIS = Swedish Standard

Table 9.0-1 (Cont'd)

Alloy		COMPOSITION (wt-%)							
		Fe	Ni	Cr	C	Co	Si	Mn	Others
AISI 304L	SIS 2352	bal.	10	19	0.03	-	0.5	1.5	
" 310	" 2361	"	20	25	0.2	-	1.0	1.5	
Nimonic 75		5.0	bal.	19.5	0.13	-	1.0	1.0	Ti 0.4
XF 818		bal.	18	18	0.2	-	0.3	0.15	Mo 7.5, Cb 0.4, B 0.7, N 0.12
12 RN 72		bal.	25	19	0.1	-	0.4	1.8	Mo 1.4, Ti 0/5, B 0.006
AISI 4130	SIS 2225	bal.	-	1.0	0.3	-	0.25	0.5	Mo 0.2
"	" 2541	"	1.4	1.4	0.4	-	0.35	0.7	Mo 0.2
	SIS 2940	"	-	1.65	0.4	-	0.35	0.65	Mo 0.3, Al 1.0
AISI 10-70-03	" 0737	"	2.0	-	3.6	-	2.0	0.7	Mg 0.06
ASTM 6351	" 4212	-	-	-	-	-	1.0	0.7	Al bal., Mg 0.9

Table 9.0-2 Composition of Alloys (wt %)

ALLOY	IRON (wt %)	NICKEL (wt %)	CHROMIUM (wt %)	COBALT (wt %)	OTHERS (wt %)
Alloys Presently Used in Automotive Stirling Engines					
Inconel 718	18.5	52.5	19.0	-	0.5 Al, 0.9 Ti, 3.0 Mo, 5.1 (Cb + Ta)
304 Stainless Steel	68.0	10.0	19.0	-	0.08 C max, 2. Mn max, 1. Si max
310 Stainless Steel	51.0	10.0	25.0	-	2. Mn max, 1.5 Si max, 0.25 C max
316 Stainless Steel	66.0	12.0	17.0	-	2. Mn max, 1. Si max, 2.5 Mo max, 0.08 C max
321 Stainless Steel	68.0	11.0	18.0	-	2. Mn max, 1. Si max, 0.08 C max, (5 x C) Ti
329 Stainless Steel	66.0	4.5	27.5	-	1.5 Mo, 0.15 Mac C
Nimonic 75	5 max	73.0	20.0	-	0.12 C, 1 max Mn, 1 max Si, 0.4 Ti
4130	97.7	-	1.0	-	0.3 C, 0.5 Mn, 0.2 Mo, 0.35 Si
7140	96.0	-	1.6	-	0.42 C, 0.55 Mn, 0.38 Mo, 0.30 Si, 1.0 Al
Alloys Proposed or Under Investigation for Application in Automotive Stirling Engines					
Inconel 617	-	55.0	22.0	12.5	9.0 Mo, 1.0 Al, 0.07 C
Inconel 625	2.5	63.0	21.5	-	9.0 Mo, 3.65 Cb, 0.2 Ti, 0.2 Al
Incoloy 800	44.0	32.5	21.0	-	0.4 Ti, 0.4 Al, 0.8 Mn, 0.5 Si, 0.05 C, 0.4 Cu
CRM-6D	64.0	5.0	22.0	-	5.0% Mn, 1.0 C, Mo, 1.0 W, 1. Cb
19-9 DL	66.0	9.0	19.0	-	1.0% Mn, 0.30 C, 1.4 Mo, 1.3 W, .4 Cb, 0.3 Ti, 0.5 Si
A-286	54.4	25.0	15.0	-	0.052 C, 1.0% Mn, 0.5 Si, 1.25 Mo, 2.1 Ti, 0.25 V, 0.25 Al

Table 9.0-3 Chemical Compositions of Alloys for Application in Automotive Stirling Engines

SECTION 10.0 PRODUCTION COST ANALYSES

10.0 PRODUCTION COST ANALYSES

10.1 Total Estimated Production Cost

The total estimated production cost of the Reference Engine, performed at United Stirling of Sweden, which was based on 200,000 units per year, was 4,878 Swedish Crowns or $1,134 (assuming a conversion rate of 4.3). The following identifies the detailed assumptions and calculations used in establishing this cost.

10.2 Basis of the Analysis

- Existing drawings.
- 200,000 engines produced per year.
- Previous available data.
  - Costs of a 2.1 liter standard production Otto engine.
  - Costs of a 2.0 liter standard production Otto engine.
  - Earlier calculations for a 75 kW Stirling engine based on 15,000 engines per year.
  - Earlier calculations for a 40 kW Stirling engine based on 100,000 engines per year.
  - A small Stirling engine based on 10,000 engines per year.
- Average Swedish salary and social costs for the first half of 1979.
- Interest at 10%.

- West German and United States machinery investment prices for 1979.
- Swedish tooling prices for the first half of 1979.
- Swedish power costs for 1979.
- Swedish statistics for maintenance.

10.3 Extent of the Analysis

- The cost analysis includes:
    - Material
    - Salary
    - Social cost
    - Tool consumption
    - Rental of buildings for production
    - Power
    - Maintenance
    - Factory overhead
    - Amortization and interest for machinery
- The cost analysis does not include:
    - Interest on work in progress (WIP)
    - Cost for storing of manufactured engines
    - Developing cost (design and prototypes)
    - Sales cost
    - Administration cost
    - Profit
    - Start-up cost

10.4 Cold Engine Parts Production

All cold parts were compared to similar parts for Otto or Diesel engines. The machinery for producing the cold parts was chosen to be the same as it would be in an engine factory situated in a highly developed, industrialized country. The following types of production were forecasted:

Crankcase	Transfer line
Bedplate	Transfer line
Crankshaft	Line production
Oil pump cover housing	Machine for complete machining
Flywheel	Line production
Gear ring, starter	Group production
Flywheel cover	Line production
Front cover	Line production
Connecting rod	Transfer line
Piston rod guide	Line production
Piston rod	Group production
Piston	Line for the lower part, group for the upper
Cylinder block	Transfer line
Liner	Line production
Guide and seats	Machine for complete machining
Water pump housing	Machine for complete machining
Impeller	Group production
Fan housing	Machine for complete machining
Fan wheel	Line production

10.5 Hot Engine Parts Production

For the hot parts of the engine, common lubrication methods were used. This means that most of the machinery and equipment will be available on the machine tool market; however, many pieces must be automated before they can be used for high production. The following were forcasted:

Gas cooler	Automatic machine for welding and cutting. Assembling machine. Vacuum furnace.
Heater tubes	Automatic welding, winding, and cutting line.
General and piston housing	Line production
Heater	Assembling machine. Vacuum furnace.
Air preheater	Cutting and bending line. Automatic resistance welding.
Assembly	In order to maintain high quality, the assembly will be station assembly. Each station will take care of one function of the engine. For instance, the heater, insulation, burner, etc. Transport between stations will be by electrically-driven carriers controlled by magnetic slings in the floor and a computer. The carrier will transport the engine on a loop from the beginning of the assembly line through testing and reject, and will be unloaded in the storage area for manufactured engines.
Testing	Test cells. Computerized test program.

10.6 Materials Cost

	Sw. cr/kg
Cast aluminum	15.00
Cast iron	2.50
Heat resistant steel sheet:	
t = 1.0	10.00
t = 0.15	20.00
t = 0.03	80.00
CRM 6D or Stellite	27.00
Normal steel sheet	2.00
Insulation	10.00

10.7 Purchased Parts

The 87% rule was used when no other costs were available.

10.8 Salary, Social Costs, Etc.

The following are valid for a new factory unit using 2 shifts per day:

	Sw. cr/h	
Salary	32.00	
Social costs 58%	18.50	
Tool Cost	17.00	
Cost for renting the building, 200 Sw cr/m^2 a year. 30 m^2/mach.	2.50	
Electricity and maintenance	4.00	
Various		2.00

Overhead	16.00
TOTAL	92.00
1 man/2 machines	67.00

10.9 Amortization and Interest Costs

Amortization plus interest at 18%, based on 3000 hours.

Investment Sw cr	Sw cr/h	Sw cr/man hour 1 man/1 machine	Sw cr/machine hour 1 man/2 machines
100,000.-	6.00	98	73
200,000.-	12.00	104	79
300,000.-	18.00	110	87
400,000.-	24.00	116	91
500,000.-	30.00	122	97
600,000.-	36.00	128	103
700,000.-	42.00	134	109
800,000.-	48.00	140	115
900,000.-	54.00	146	121
1,000,000.-	60.00	152	127

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16. Abstract

This report describes the Reference Stirling Engine System, which was to provide the best possible fuel economy while meeting or exceeding all other program objectives. It was designed to meet the requirements of a Reference Vehicle, which is a 1984 GM Pontiac Phoenix (X-body). This design utilizes all new technology that can reasonably be expected to be developed by 1984 and that is judged to provide significant improvement, relative to development risk and cost.

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